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NOTICE

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20. ABSTRACT (Continue on reverse side if necessary and identify by block number)

The report describes the continued development and test verification of digital computer models used to simulate hydraulic systems under dynamic conditions.

A hydraulic acoustic generator was built and tested for frequency response work. Pulsation attenuation techniques are presented and several modifications were made to the Hydraulic Frequency Response Computer Program (HSFR) including a standing wave plot option.

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20. Abstract (Continued)

An empirical time domain pump model was developed and additional verification and development of the piston pump transient model was accomplished. Several enhancements were made to the Hydraulic Transient Analysis Computer Program (HYTRAN).

A quasi-transient capability was added to the Steady State Flow Analysis Computer Program (SSFAN). The SSFAN Program was updated to extend the calculation capability, add new component models, program features, and reduce the program size.

Several programming features were added to the Hydraulic Transient Thermal Analysis Computer Program (HYTTHA) including a temperature default section and environment section. Further verification work was accomplished on the HYTTHA program.

Development and verification of a Hydraulic Line Mechanical Response (HLMR) computer program for predicting line mechanical response due to predicted pump pressure pulsations has been partially achieved. Using the NASTRAN finite element model computer program was also investigated.

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PREFACE

The final report was prepared by the McDonnell Aircraft Company, Design Engineering Power and Fluid System Department, McDonnell Douglas Corporation under contract F33615-78-C-2026, with supplemental agreement P0003.

The effort was sponsored by the Aero Propulsion Laboratory, Air Force Systems Command, Wright-Patterson AFB, Ohio under Project No. 3145-3027 with AFWAL/POOS, and was under the direction of Paul Lindquist and William Kinzig.

The final report covers work conducted from 1 June 1978 through 31 January 1980. At McDonnell, Neil Pierce directed the program and was Principal Investigator. Special acknowledgement is also given to J. B. Greene, R. J. Levek, H. DeGarcia, R. F. Deshazer, R. E. Young, M. J. Stevens, L. E. Clements and M. A. Furman.

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TABLE OF CONTENTS

TABLE OF CONTENTS (continued)

TABLE OF CONTENTS (continued)

TABLE OF CONTENTS (Continued)

TABLE OF CONTENTS (Continued)

LIST OF ILLUSTRATIONS

FIGURE	LIST OF ILLUSTRATIONS (Continued)	PAGE

FIGURE	LIST OF ILLUSTRATIONS (Continued)	PAGE

LIST OF ILLUSTRATIONS (Continued)

LIST OF ILLUSTRATIONS

LIST OF ILLUSTRATIONS

LIST OF ILLUSTRATIONS

LIST OF TABLES

LIST OF TABLES

SECTION I

INTRODUCTION

This report describes work performed under the Advanced Fluid System Simulation (AFSS) contract. The purpose of the AFSS program is to establish and validate the modeling required for a comprehensive dynamic simulation of aircraft hydraulic systems. Prior work in this area has been documented in References (1) and (2). The computer programs are documented in References (3) through (10).

The AFSS tasks involved improvement, development and verification by test of digital computer models and/or programs in five areas.

1. HYDRAULIC TRANSIENT ANALYSIS (HYTRAN) COMPUTER PROGRAM DEVELOPMENT

The HYTRAN program can provide useful predictions of peak waterhammer pressures and system response to changing flow demands. HYTRAN general purpose math models are adequately verified so that they can be used with confidence to analyze hydraulic systems. They can be used to write a simple system model or they can be modified to create very detailed models to predict actual system and component performance.

HYTRAN is a complex computer program. The user must be familiar with transient phenomena in hydraulic systems and know how to interpret program results. Unacceptable results may be due to inaccurate input data or unrealistic design conditions.

Further HYTRAN development involved increasing the useability of this complex tool. An empirical pump model was written, further verification work was performed on the basic pump model, a lossless line model was developed and several program improvements were made. The above items are detailed in this report.

2. HYDRAULIC SYSTEM FREQUENCY RESPONSE (HSFR) COMPUTER PROGRAM DEVELOPMENT

The HSFR program predicts how oscillatory flows and pressures caused by the acoustical energy content of a pump output are transmitted through the lines and components of a hydraulic system. The program predicts the pump speeds at which major resonances occur, and defines the amplitude and location of the oscillatory pressure and flow standing waves. The simulated system input data can be easily changed to investigate various practical system modifications for the attenuation and/or relocation of the major resonant conditions.

The HSFR program can provide sufficiently reliable predictions of resonant frequencies. However, the accuracy of predicted pulsation amplitudes at resonant pump speeds varies widely.

The final report discusses attempts to improve the accuracy of the pulsation amplitude predictions by test data attained using a single piston hydraulic acoustic source.

Various pulsation attenuation techniques are presented and organized to provide guidance in solving attenuation problems.

Several program improvements are documented including correction of the pump flow calculation and a standing wave plot option.

3. STEADY STATE FLOW ANALYSIS (SSFAN) COMPUTER PROGRAM DEVELOPMENT

The SSFAN program was developed to predict steady state flows and pressures in a closed loop Hydraulic System. It uses a building block approach so that new elements or components can be added with minimum change to the program. A matrix method is used to compute steady state flows throughout the system line network. SSFAN corrects viscosities for pressure, determines whether flow is laminar, transitive or turbulent to apply appropriate resistance factors.

A Quasi-transient capability and several program improvements including a flow regulator model and a new matrix calculation technique are discussed in this report.

4. HYDRAULIC TRANSIENT THERMAL ANALYSIS (HYTTHA) COMPUTER PROGRAM DEVELOPMENT

The HYTTHA Program predicts the effects of system heat generation and dissipation of the temperature and performance of an aircraft hydraulic system. The program can simulate closed loop systems and calculates flows, pressures state variables, component temperatures, fluid temperatures, and line wall temperatures throughout the system.

HYTTHA was written to analyze temperature phenomena that are particularily complex. Consequently, large amounts of input data are required.

Several improvements to increase the program useability and further verification tests are discussed in this report.

5. HLMR PROGRAM DEVELOPMENT

Development and verification of a computer program for predicting line mechanical response due to predicted pump pulsations has been achieved, though it is somewhat limited. The program can be used to determine the mode shape and frequency of fundamental line responses. Higher modes of line response, which were the predominant responses of the test specimens conducted to date, can be predicted for certain configurations. However, these solutions could not be generalized and therefore a general purpose Hydraulic Line Mechanical Response (HLMR) computer program is still in the formative stages.

Tests have been conducted on one-size diameter specimens using accelerometers and strain gages as transducers. Each hydraulic resonance caused a significant mechanical response within the operating regime of the pump. Reduction of pump pulsation energy produces a reduction in line response. Development and use of effective wide band pulsation alternatives offers an attractive and perhaps more cost effective near term alternative than the development of a general purpose HLMR program which should be a long term endeavor.

SECTION II
HYTRAN PROGRAM

1. BACKGROUND

The HYTRAN Program simulates the dynamic response of a hydraulic system to sudden changes in load flow demand. HYTRAN is a computer program requiring the user to be familiar with transient phenomena in fluid systems. Unacceptable simulations often result from inaccurate input data or unrealistic design conditions.

The statement of work emphasized improving and increasing program useability.

An empirical pump model was developed and verified with test data. Further work on the full pump model defined input parameters and achieved better test data correlation. The feasibility of incorporating a lossless line model was studied and several improvements were made to the HYTRAN Program. The above items are discussed in the following sections.

2. HYTRAN EMPIRICAL PUMP MODEL

The original HYTRAN pump model was written to study pump/system stability under dynamic loading conditions. The model yields acceptable correlation with test data at the expense of the time required to measure and derive pump design parameters for the full pump model. However, all the detailed data required for the HYTRAN pump model is not available from pump manufacturer data sheets. Consequently, the HYTRAN program user must either disassemble the pump or ask the supplier for the information. Oftentimes the data is proprietary and may not be released without a special arrangement. The entire process can be slow and cumbersome for the HYTRAN user. The designer must guess at many performance parameters.

Two HYTRAN empirical pump models have been developed to overcome the HYTRAN full pump model detailed data requirement. Input data consists of parameters measured directly from test data. Furthermore two or three standard systems can be tested which provide a spectrum acceptable for covering present system usage.

The following sections describe the pump models, the steady state and transient tests, and model correlation with test data. The final section presents a procedure outline for usage of each empirical pump model.

a. Empirical Pump Model Development

The first step in developing a variable displacement pressure compensated hydraulic pump empirical model was to evaluate the data that can readily be obtained either from testing the unit or from the manufacturer's specification sheets. Steady state pump performance is usually provided or can be easily measured on a test stand. Transient data should be obtained from a circuit that adequately matches the actual operating system. If this cannot be accomplished, knowledge of the pumps transient performance in various test systems will provide data that can be extrapolated to the actual system.

A variable displacement pressure compensated hydraulic pump, responds to variations in system pressure due to load perturbations. When system pressure changes from the pumps compensator setting, a change in pump output flow results. The approach taken in developing the HYTRAN empirical pump model was to establish an algorithm that computes pump flows given a change in system pressure.

Because of the program constraints many of the parameters used to describe the complex pump/system interaction cannot be measured directly but must be inferred from test data. The pump transient response program can be simplified by considering only pressure side dynamics. The HYTRAN program further requires the pump model to be separated from the line dynamics. Although this simplifies the pump model mechanics, it complicates the problem of requiring the model to define the pump interaction with any system configuration. Without knowledge of the internal parameters that define pump operation, the model input parameters will change from system to system. This is not necessary for the existing HYTRAN pump model because the critical internal pump parameters are defined and the interrelationships are modeled.

(1) Evaluating Pump Performance Data

Pump steady state performance in a system can be defined by several data curves. Pump output values at any speed is represented by a flow - pressure curve. The pressure drop between case and suction versus case drain flow defines the pumps leakage characteristics in a particular system but does not generally define the pumps internal leakage flows from outlet to case. This leakage value could be obtained from the flow pressure curve. The case flow is then subtracted from the total leakage for the leakage flow back to the inlet.

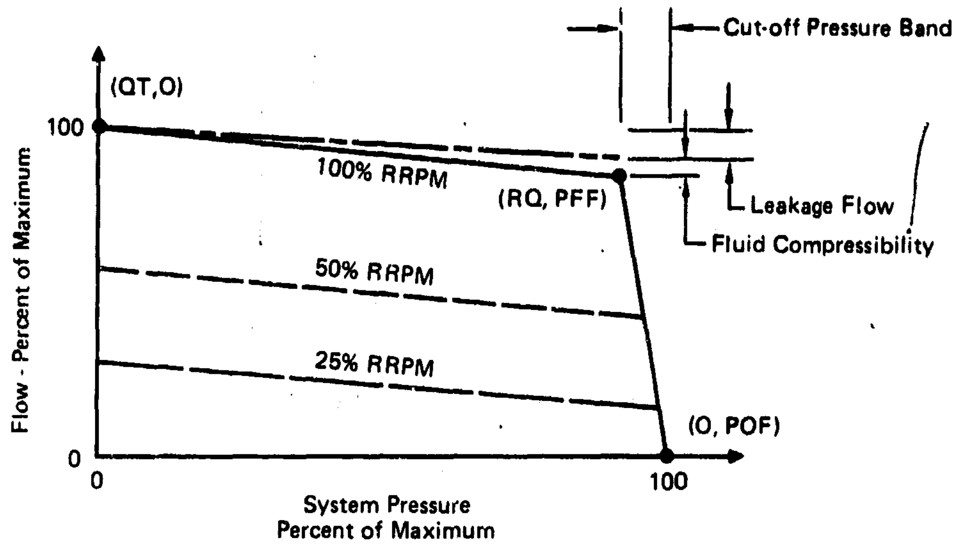
Pumps are typically designed to operate with a minimum inlet pressure to preclude cavitation and subsequent erosion of critical pumping surfaces. Below the minimum inlet pressure pump outlet flow decreases linearly with the inlet pressure.

Figure 1 presents a typical variable displacement pressure compensated performance curve. The boundaries of the curve are defined by measured values which determine the operating characteristics. When system pressure rises due to a load, the pump output flow remains constant until the pressure reaches the pressure setting of the compensator valve spring. A further increase in system pressure moves the compensator valve spool and ports flow to the hanger actuating cylinder. The velocity of the hanger is therefore proportional to the position of the compensator valve. The rate of pump flow reduction varies with the pressure overshoots. The hanger angle is reduced until the steady state flow is compatible with the regulated system pressure which the pump is designed to provide. The regulated system pressure is defined in terms of a bandwidth as shown in Figure 1.

The flat cut-off compensator holds a nearly constant pressure under all steady state conditions. Faster response speed with less overshoot can be achieved with the flat cut-off type pump.

Normally the pump cannot maintain an entirely flat characteristic. A 3 to 5% variation in outlet pressure does take place. The bandwidth occurs because of pump leakage past the hanger actuating cylinder requiring larger supply pressures to provide enough flow to maintain hanger position and pump control valve leakage. Figure 1 shows that at the lower flow rates (smaller hanger angle) the supply pressure is greater. Therefore, the actuator leakage is larger with more actuator displacement.

When the load is decreased, system pressure falls and the hanger spring will move the hanger to increase pump outlet flow. Sufficient flow will be provided until a pressure in the cut-off band is reached.



RQ - Rated outlet flow
QT - Total flow (rated flow + leakage flow)
PFF - Rated pressure (full flow)
POF - Rated pressure (zero flow)
RRPM - Rated RPM

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FIGURE 1 VARIABLE DISPLACEMENT PRESSURE COMPENSATED PUMP OPERATING CHARACTISTICS

Pump leakage paths to case are generally long and can be treated as a linear function with pressure. Then the leakage flow gradient is the slope of the flow versus pressure curve for a pressure value less than or equal to the minimum regulated value.

Since pump output flow is directly proportional to pump speed, other speeds will be parallel to the rated RPM curve of the pump. This is illustrated in Figure 1 by the dashed lines at 50% and 25% of rated pump speed.

The pump is a highly underdamped servomechanism whose transient response is dependent on many pump and system parameters. The block diagram in Figure 2 illustrates the interdependency of the pump and system. The pump provides flow for the system load. When the load flow changes, so does system pressure. The perturbations in line pressure are sensed by the pump and the pump outlet flow is adjusted. The change in flow depends on how far the line pressure is away from a reference value set in the pump.

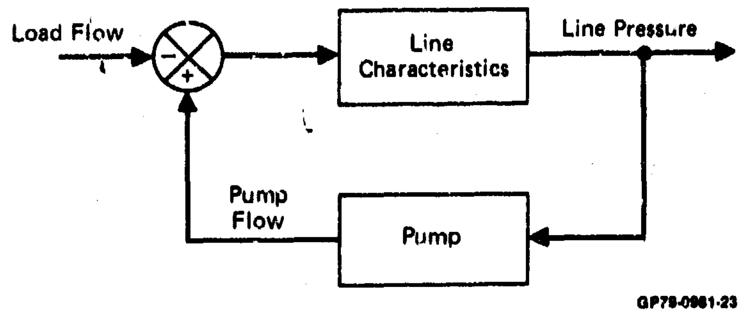

FIGURE 2 PUMP/SYSTEM BLOCK DIAGRAM

Pump transient performance depends on factors which are critical to pump/system stability. To isolate pump effects from system effects in a test stand is a difficult task when critical pump parameters are not monitored during the transient. The first task is to list the measurable pump/system parameters and evaluate the information they can provide.

Pump port pressures are easily measured and give a time history of pump and system interaction due to a load change. An opening or closing valve generates a step input into the system which results in the classic waterhammer wave. For a line system with a reservoir at one end and a valve at the other the pressure response to a valve input will oscillate at a frequency defined by equation (1).

$$f = \frac{c}{4\ell} \qquad (1)$$

Where

c = speed of sound in fluid (in/sec)
l = line length (in)
f = oscillation frequency

Equation (1) can only be used if the valve closes in less than the critical closing time (t_c).

$$t_c = \frac{2\ell}{c} \qquad (2)$$

When a pump replaces the reservoir in the line/valve system, the frequency (f) is altered. The difference is due to pump system interaction. The pump effects to the pressure signal can be readily observed. However, if the valve closing time is greater than t_c, return system dynamics further complicate the data reduction.

For a particular system configuration, time constants can be measured resulting from turn-on and turn-off transients at various pump flow rates. The time constants reflect viscous line dissipation, fluid bulk modulus effects, line frictional losses, and pump response.

The mechanical torque can be measured and converted to hydraulic power. Torque can provide an approximation to pump flow response for turn-off transients and turn-on transients less than full flow. (No accurate rate information can be obtained unless hanger position is known).

The control flow rate depends on the difference in pressure from the set point. The hanger actuator integrates the flow so the actuator velocity is proportional to the error in pressure. Consequently the rate of change of flow varies with input pressure.

Pump case pressure and inlet pressure will mirror hanger response. No rate data can be obtained from the inlet pressure because of interference from return system dynamics and the usually noisy suction pressure signal measurements. Control pressure would provide better rate information.

Pump transient response is influenced by hydraulic system characteristics and the pump responds to pressure and not flow. Pressure builds up or decays as a function of pump displacement, system volume and system impedance. Actual pressure recovery of the system resulting from changes in flow demand depend on system constants and pump characteristics. It is not possible to stipulate the response of a pump in a system unless all these constants are known or can be determined by testing.

(2) Algorithm Development

Pump mathematical models assume many forms. A particular model is usually written to fit a set of conditions. The simplest model typically requires parameters integral to pump design as control valve gain, pump flow gain, and internal leakage characteristics. Because the information on any pump is limited to pressure and torque data in various systems, the empirical pump model by design must be kept simple.

In deriving the simplified model, inherent physical properties of the pump must be ignored. Non-linearities and distributed parameters which are present in the real pump must be estimated by a linear lumped parameter model. This will result in ordinary differential equations which contain constant coefficients.

To describe the input and output relationship of the pump a transfer function can be developed. The transfer function concept only applies to linear time invariant systems.

A simple transfer function for the pump in Figure 3 relates the Laplace transform of the pump outlet flow to the Laplace transform of system pressure.

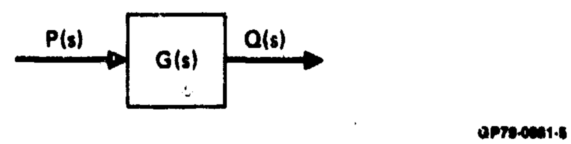

FIGURE 3 SIMPLE PUMP TRANSFER FUNCTION BLOCK DIAGRAM

The transfer function if written as:

$$G(s) = \frac{Q(s)}{P(s)} \tag{3}$$

and is a property of the pump itself, independent of the input driving function. The Laplace transform technique allows the pump internal dynamics to be represented by algebraic equations in the S-Plane. The highest power of s in the denominator of the transfer function determines the highest derivative or system order.

An equivalent block diagram of the pump and load system is shown in Figure 4.

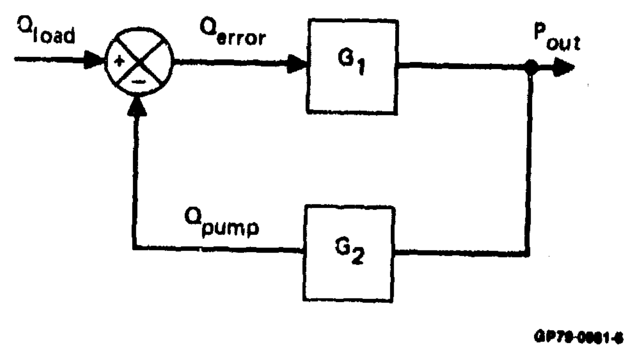

FIGURE 4 PUMP AND LOAD SYSTEM BLOCK DIAGRAM

The G_1 box represents the description of the line and load dynamics. The G_2 function describes the pump operation. The HYTRAN program simulates the entire system. The line and load dynamics are interrelated through the valve and line models. The pump model closes the loop in the simulation. The task is to write a transfer function to adequately predict the pump response in any system.

The line and load dynamics are in the same loop as the pump. Any pump outlet pressure and outlet flow data obtained in a test stand would result in a transfer function defining the pumps closed loop characteristics. Deriving the transfer function would be a detailed task. The input forcing function is a complicated pressure function, and the torque measurement can only provide some degree of flow response without knowing the actual flows.

A step response to the closed loop system is provided by a control valve with an operating time less than t_c. If the line volume is sufficiently small the line characteristics can be ignored. An approximation of the pumps response with a constant in the feedback loop can be measured.

Two empirical pump models were written. The first model represents a first order system that describes the pump/system dynamics interaction. The other model relies on a second order relationship to define the pump/system transients.

(a) First Order Pump Model

A pump/system flow model was written assuming a linearized system where variable changes are kept small. The pump flow minus the load flow in a small time interval dt is equal to the change in pressure times the change in stored fluid necessary to cause a unit change in pressure. A differential expression is written as

$$(Q_{in} - Q_{out})dt = C\, dP \tag{4}$$

Where

$C = \frac{A\ell}{\beta}$

A = Line Area (in^2)

ℓ = Line Length (in)

β = Fluid Bulk Modulus (PSI)

Q_{in} = Pump Flow (CIS)

Q_{out} = Load Flow (CIS)

P = System Pressure (PSI)

The system provides a resistance to the flow Q_{out} at pressure P. The relationship between Q_{out} and P is

$$Q_{out} = \frac{P}{R} \tag{5}$$

If R and C are constant, a differential equation for a fluid flow system, combining equations (4) and (5), is

$$RC\frac{dP}{dt} + P = R\, Q_{in} \tag{6}$$

Taking the Laplace transform of Equation (6) assuming zero initial conditions yields

$$RCs\, P(s) + P(s) = R\, Q_{in}(s) \tag{7}$$

If Q_{IN} is the input and P the output the system transfer function is

$$\frac{P(s)}{Q_{IN}(s)} = \frac{R}{RCs+1} \tag{8}$$

However, the outlet pressure can be described in terms of system resistance (equation (5)). The transfer function is then

$$\frac{Q_{out}(s)}{Q_{in}(s)} = \frac{1}{RCs+1} \tag{9}$$

The input flow can be described by the pressure/flow characteristics of the pump. The output flow will be a first order response. A simplified block diagram is shown in Figure 5.

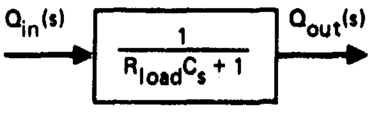

FIGURE 5 FIRST ORDER PUMP/SYSTEM BLOCK DIAGRAM

A unit step response can be used to simulate a valve closure in a hydraulic system. The Laplace transform of the unit-step function is 1/s. Substituting $Q_{in}(s) = 1/s$ into equation (9) gives

$$Q_{out}(s) = \frac{1}{R_{load}C+1} * \frac{1}{s} \qquad (10)$$

Expanding into partial fractions and setting $T = R_{load}C$

$$Q_{out}(s) = \frac{1}{s} - \frac{T}{Ts+1} \qquad (11)$$

The inverse Laplace transform of (11) yields

$$Q_{out}(t) = 1 - e^{-t/T} \qquad (12)$$

When t is equal to T, $Q_{out}(t)$ is equal to .632 or the response $Q_{out}(t)$ has reached 63.2% of its final value at time T.

T is called the time constant of the system. The smaller the time constant, the faster the system response. It can be easily shown that after four time constants the response curve is within 2% of the final value.

The first order flow response can provide adequate predictions of pump flow in systems where the flow response is less than 50 HZ. The system response time can be used to approximate the flow response.

The transfer function in equation (9) can be used to build a HYTRAN pump model. A block diagram of the model is shown in Figure 6.

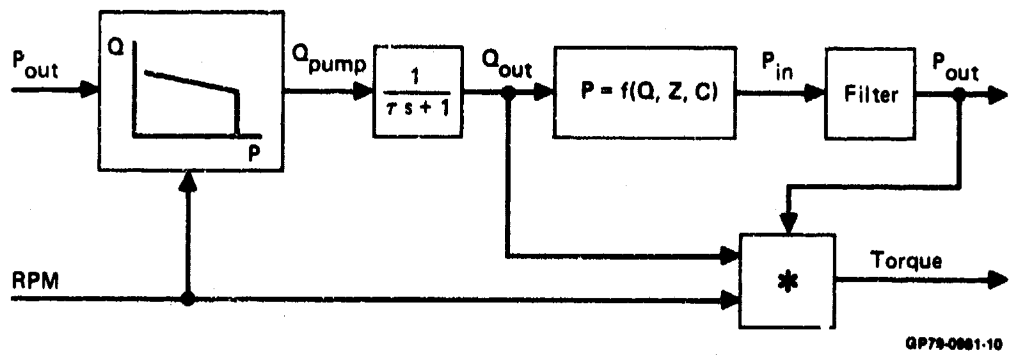

FIGURE 6 HYTRAN FIRST ORDER EMPIRICAL PUMP MODEL

System line pressure is used to determine pump outlet flow from the steady state pressure/flow characteristics. The flow (QPUMP) then passes through the first order response network. The resulting pump flow (Q_{out}) is combined with the line dynamics to compute a pump outlet pressure. The pressure is passed through a filter circuit with a small time constant to eliminate any high frequency pressure spikes. The pump outlet pressure and flow is combined with the pump operating RPM to obtain a torque value. If the calculation time interval is sufficiently small, the model can provide a good approximation of pump flow and pressure performance in a system.

(b) Second Order Pump Model

A block diagram of a second order system is shown in Figure 7.

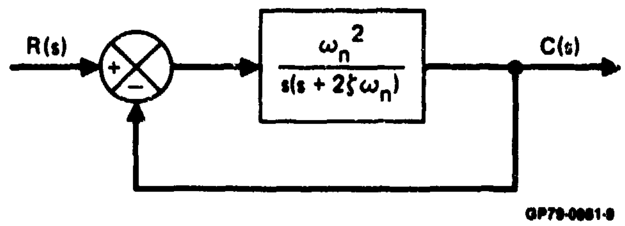

FIGURE 7 SECOND ORDER BLOCK DIAGRAM

The closed loop transfer function $\frac{C(s)}{R(s)}$ can be written

$$\frac{C(s)}{R(s)} = \frac{\omega_n^2}{s^2 + 2\zeta\omega_n s + \omega_n^2} \tag{13}$$

The dynamic behavior of a second order system can be described in terms of two parameters, ω_n - the system natural frequency and, ζ - the damping coefficient. If ζ is between 0 and 1 the closed loop poles are complex conjugates and lie in the left hand s plane. The damping rate depends on how far the poles lie from the real axis.

If R(s) is a unit step input, equation (13) can be written

$$C(s) = \frac{\omega_n^2}{s^2 + 2\zeta\omega_n s + \omega_n^2} \cdot \frac{1}{s} \tag{14}$$

Equation (14) can be expanded as

$$C(s) = \frac{s^2 + 2\zeta\omega_n s + \omega_n^2 - (s^2 + 2\zeta\omega_n s)}{(s^2 + 2\zeta\omega_n s + \omega_n^2)(s)} \tag{15}$$

$$= \frac{1}{s} - \frac{s + 2\zeta\omega_n}{s^2 + 2\zeta\omega_n s + \omega_n^2}$$

Defining

$$\omega_d^2 = \omega_n^2 (1 - \zeta^2)$$

Equation (15) can further be reduced

$$C(s) = \frac{1}{s} - \frac{s + \zeta\omega_n}{(s + \zeta\omega_n)^2 + \omega_d^2} - \frac{\zeta\omega_n}{(s + \zeta\omega_n)^2 + \omega_d} \tag{16}$$

Taking the Laplace transform of equation (16) yields

$$c(t) = 1 - \frac{e^{-\zeta\omega_n t}}{\sqrt{1 - \zeta^2}} \sin\left(\omega_d t + \tan^{-1}\frac{\sqrt{1 - \zeta^2}}{\zeta}\right) \tag{17}$$

Where $t \geq 0$

When ζ is equal to zero, the response is undamped and the oscillation continues indefinitely. A damping coefficient greater than or equal to one results in an undefined response. Equation (17) can be approximated and solved for the critically ($\zeta=1$) and overdamped cases ($\zeta>1$). The response is similar to that of a first order system.

Equation (17) is used to compute the time response of the pump outlet flow.

In the derivation of the second order response it is assumed that the transient control valve operation can be approximated by a step input, and the valve completely closes off the line. This is usually not the case in practical hydraulic design. Oil leaks past the valve and energy dissipation rises. The net result are changes in the effective natural frequency and damping of the system. The effect on frequency is considerable and that on damping can be by a factor of ten. Calculations of system performance for one operating condition in a system may be in error for other conditions. This is due mainly to the non-linearity of fluid power systems and the care in selecting system variables cannot be overemphasized.

The second order model more closely emulates the mechanical action of the pump than the hypothetical first order model. Variable displacement pumps usually incorporate a control valve, which strokes a piston, that moves the pump hanger, changing the output flowrate. All these functions including the feedback pressure can be broken down into individual functions, then pieced together to form a model.

The pump controlling compensator valve will initially sense a change in system pressure. A definite lag time is involved until the compensator starts to move and ports fluid to or from the hanger actuator. The response of the control spool is important for it controls the rate at which the hanger will respond and eventually dampen to a final flow value. The delay time can be easily measured as the time the output pressure changes to the time that the torque changes.

The control spool position and the hanger actuator position can be related through a second order response. The actuator position is integrated to obtain a velocity which moves the hanger to a pump flow. The flow reacts with the system pressure providing a feedback through the compensator network.

In the pump model the second order response was used to simulate the control spool and hanger dynamics. The transfer function output was multiplied by a proportional controller to obtain the outlet flow. Feedback to the second order transfer function was provided by outlet pressure referenced to the final steady state pressure value.

Damping and frequency values for the second order transfer function are obtained from the transient torque characteristics. Steady state pump pressure flow characteristics are used to determine the pump model final steady state values.

A schematic diagram of the pump model is found in Figure 8. The model has no pressure feedback loop to the control valve. Once the transient is started the second order function will propogate at the selected natural frequency and damping rate. A pressure reference is provided by the compressibility flow (Q_{comp}). The flow is combined with the pump outlet flow as a correction factor. The resultant flow is then solved with the line boundary equation to obtain a pump outlet pressure.

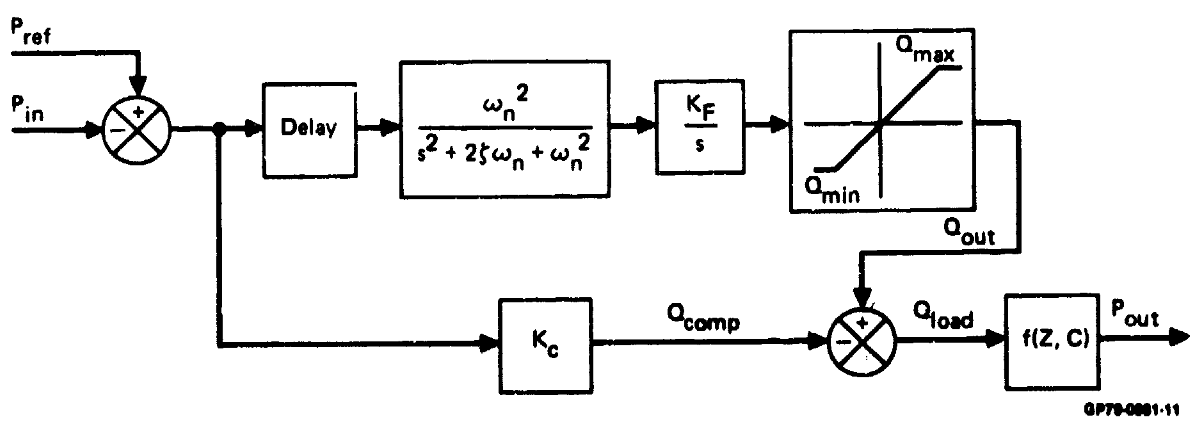

FIGURE 8 HYTRAN SECOND ORDER EMPIRICAL PUMP MODEL

It was found that the first order model was inadequate in small volume systems. A second order function did provide a stable response characteristic.

b. Hydraulic Pump Tests and Test Set-up

The objective was to test two variable displacement pressure compensated hydraulic pumps to determine their steady state and transient performance in different systems. The testing would provide input data for developing an empirical pump model and help verify the pump model.

Two pumps were tested in the Hydraulic Performance Analysis Facility (HPAF), a F-15 hydraulic pump (ABEX, 56 GPM @ 4600 RPM, 3050 psi) and a F-4 hydraulic pump (ABEX - AP10V-62,25 GPM @ 3250 RPM, 3000 psi). Each pump was tested in three different systems.

The basic configuration for the pump tests is shown in Figure 9. The outlet line lengths were changed to provide three system sizes. The line from the reservoir to the pump inlet was 25" long to preclude pump cavitation during the transient and steady state test series.

For the F-15 pump tests the line lengths were 58.5", 452.0" and 887.75" of 1"O.D. x .058" wall stainless steel tubing. Accounting for the pump manifold volume, the three test system volumes between the pump outlet and transient load valve were 37.27 in.^{3}, 278 79 in.^{3} and 546.23 in.^{3}. The F-15 pump installation is shown in Figure 10.

The line lengths for the F-4 pump tests were 51.0", 441.0" and 876.75" of 1"O.D. x .058" wall stainless steel tubing. System volumes between the F-4 pump and the transient control valve were 31.3 in.^{3}, 270.66 in.^{3} and 538.1 in.^{3}. The F-4 pump installation is shown in Figure 11.

The transient valve was a Marotta Valve Corporation, 0-3000 psi, 2 way -2 position solenoid operated shut-off valve (P/N 205883-1). The unit had 3/8" ports with a maximum flow rate of 110 CIS at 3000 psi. The valve operating time was approximately 10 milliseconds for turn-off transients and up to 30 milliseconds for turn-on transients.

The F-15 Jet Fuel Start (JFS) Accumulator had a shut-off valve on the inlet and a plugged return line. The gas precharge pressure was 1500 psig. Figure 12 shows the installation for the JFS Accululator, transient control valve, and bypass line which provided a system steady state leakage flow.

MIL-H-5606C hydraulic fluid was used in the test set-up. The dissolved air content of the fluid was typically 12% by volume.

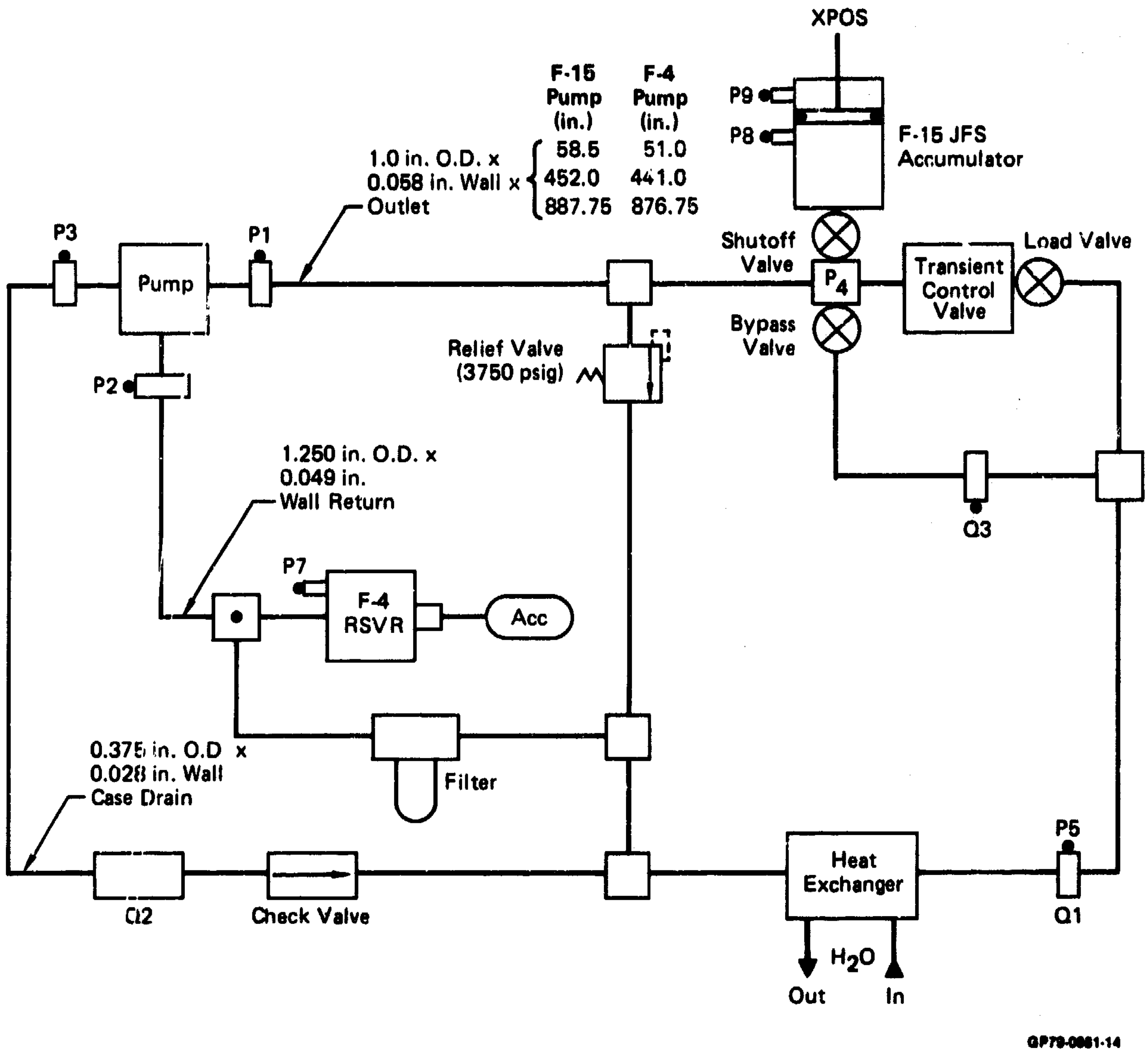

FIGURE 9 HYTRAN EMPIRICAL PUMP MODEL VERIFICATION TEST STAND

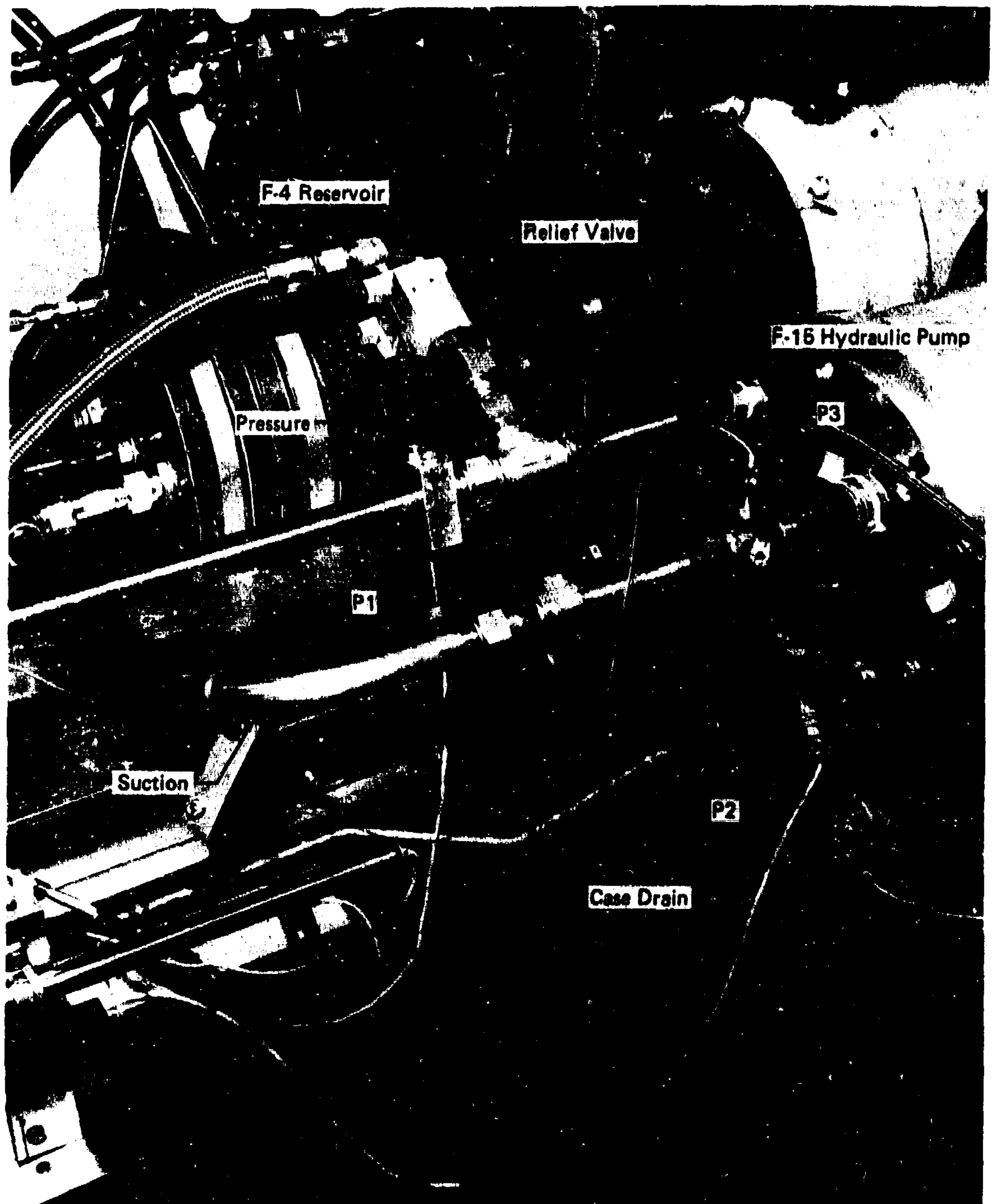

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FIGURE 10 F-15 HYDRAULIC PUMP INSTALLATION

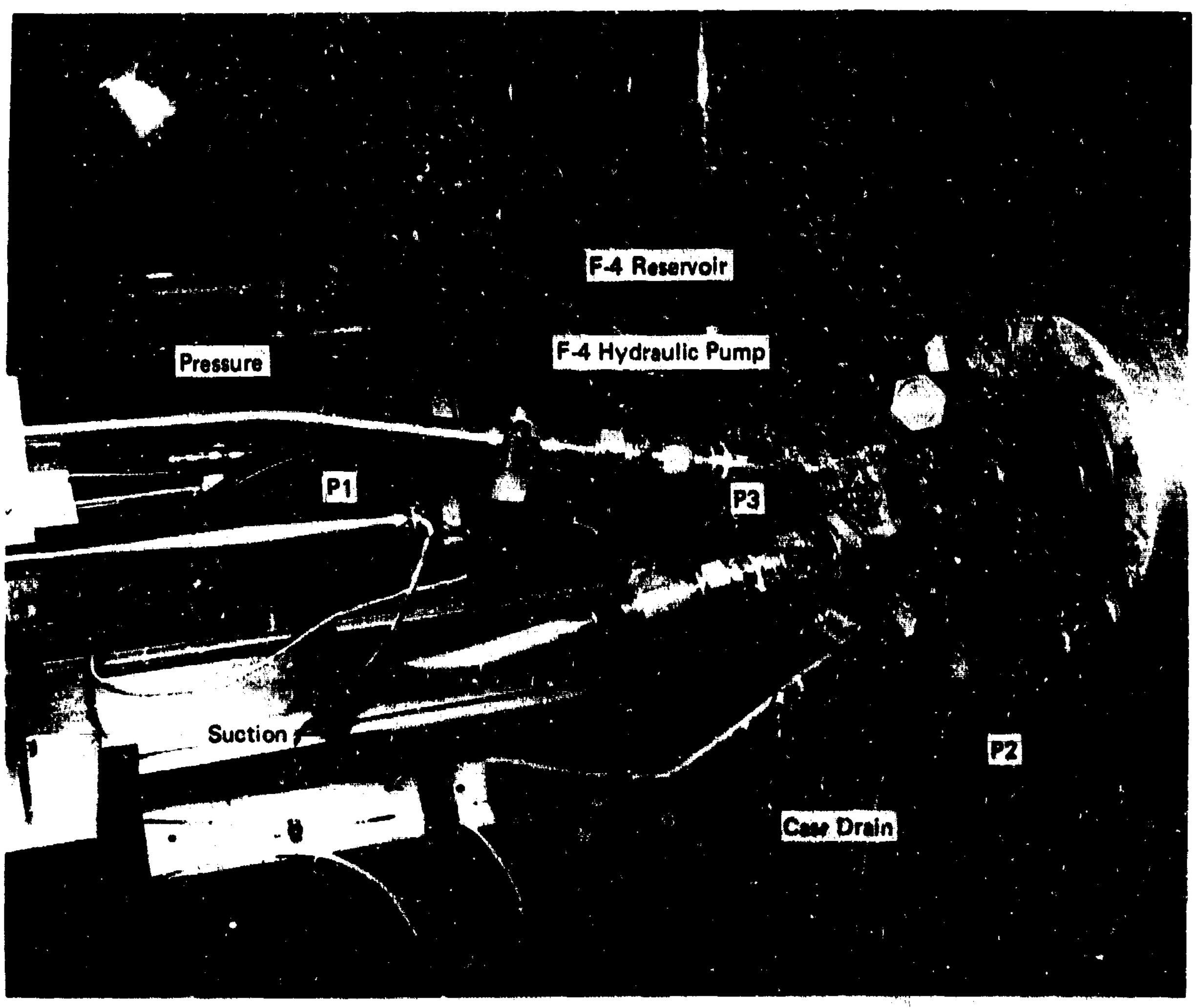

FIGURE 11 F-4 HYDRAULIC PUMP INSTALLATION

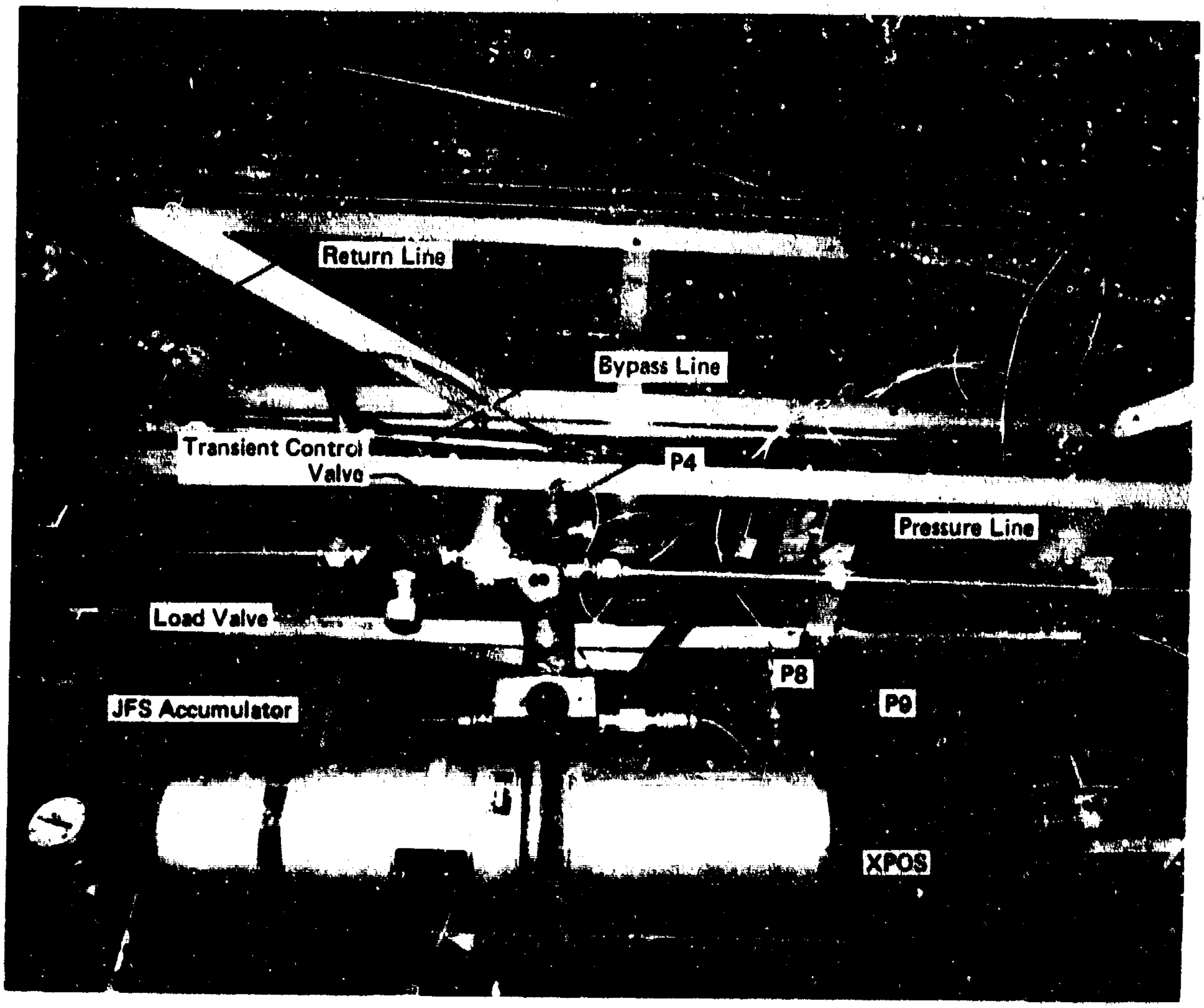

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FIGURE 12 EMPIRICAL PUMP TEST LOAD INSTALLATION

The following parameters were measured from the test stand:

PUMP OUTLET PRESSURE (P1)

PUMP OUTLET FLOW (Q1) - STEADY STATE ONLY

PUMP CASE PRESSURE (P2)

PUMP CASE FLOW (Q2) - STEADY STATE ONLY

PUMP INLET PRESSURE (P3)

RESERVOIR PRESSURE (P7)

TRANSIENT CONTROL VALVE PRESSURE (P4)

BYPASS FLOW (Q3)

DOWNSTREAM LOAD VALVE PRESSURE (P5)

TRANSIENT CONTROL VALVE CURRENT (XV)

JFS ACCUMULATOR PISTON POSITION (XPOS)

JFS ACCUMULATOR OIL PRESSURE (P8)

JFS ACCUMULATOR GAS PRESSURE (P9)

DRIVE TORQUE (DT)

NOTE: The instrumentation locations in Figures 9, 10, 11 and 12 will be referred to by the parameter number as listed above throughout the remainder of the HYTRAN Empirical Pump sections. We suggest the reader make a copy of the system schematic in Figure 9 for further reference.

Pump characteristics and performance are specified to be compatible with the aircraft hydraulic system. Because each pump is designed to operate in a particular system, it was necessary to determine whether a unit was stable in the test system. Stability refers to the freedom from oscillation of the delivery control compensator mechanism. All pumps under test operating conditions must recover steady state operation after initial response to a change in flow demand and have a steady operation down to 1000 RPM.

Prior to transient testing, a search was made for pulsation resonances in each test circuit to preclude damaging the system or pump. The pump speed was varied from 1000 to the rated RPM. Table 1 presents a list of acceptable pump operating speeds based on the frequency sweeps.

TABLE 1 PUMP OPERATING SPEEDS

PUMP	SYSTEM VOLUME (IN^3)	SPEED (RPM)	NOTES
F-15	278.	3750	200 psi p-p pulsations (400 psi p-p @ 4600 RPM)
F-15	546.	3750	210 psi p-p pulsations
F-15	37.	1950	200 psi p-p pulsations compensator was unstable above 2400 RPM (1000 psi p-p @ 2400 RPM)
F-4	31.3	3750	No stability problems in the short system
F-4	270.6	3750	
F-4	538.1	3750	

The first system investigated had a 278 in^3 volume. On the test stand the F-15 instrumented pump case pressure was about 60 psig with 77 CIS outlet flow and 55 psig reservoir pressure. System leakage of 2.0 CIS was provided by a bypass line around the transient control valve (Figure 9).

Generating a turn-off transient either with a hand valve or the transient solenoid valve, caused the pump compensator spool to oscillate. Pump outlet pressures would vary ± 200 psi. Raising the bypass leakage flow to 14 CIS would stop the oscillations, but the higher flow rate was not desirable because of the reduced pump hanger response. The leakage flow was reset to 2.0 CIS and the case pressure was increased to 85 psig by a hand valve in the case drain line. The compensator oscillations would dampen within two seconds for turn-off transients. Increasing the bypass flow or case pressure would decrease the turn-off transient damping time. Varying pump rpm would also affect the pump stability.

Several other F-15 pumps were tried in the test circuit. They exhibited the same operating characteristics.

For a turn-off transient, the pump actuator must stroke the hanger to the minimum flow condition. The oil to fill the actuator comes from the system. Consequently, system dynamics are an integral part of the pump response. Another key factor for pump stability is case pressure which acts on the barrel, hanger and actuator. Increasing the case pressure improved the damping characteristics. Higher case pressure reduced case outlet flow and increased the leakage flow back to the inlet. The resultant higher inlet pressure can contribute to better dynamic pump stability. Raising the case pressure above 85 psig did not appreciably change the pump damping characteristics.

No stability problems were encountered for turn-on transients. The system dynamics are essentially isolated from the pump actuator through the compensator valve. The actuator flow dumps directly to case as the pump is destroking to the selected flowrate.

When the load flow is zero the pumps pressure-flow leakage will increase the pump/system loop gain and the line time constant. At rated flow the line pressure flow curve will decrease the pump/system loop gain and line time constant.

The transient test series on the 278 in^3 system was completed with 2.0 cis outlet flow and 85 psig pump case pressure. During steady state testing the F-15 instrumented pump failed. The pump was running at 215.6 cis and rated rpm (4600) with reservoir pressure at 55 psig. The pump speed was increased to 4800 rpm.

Pump outlet flow was approximately 223 cis when the speed was decreased to 4600 rpm the case pressure began to oscillate between 75 and 300 psig. The pump speed was reduced to 2300 rpm and the case pressure stabilized at 125 psig. The main flow control valve on the pump outlet was closed in an attempt to reduce the pump outlet flow. As the flow decreased the case temperature rose to 260°F in less than one minute. The drive was then shut off.

The pump was allowed to cool to 180°F before a restart was attempted. On restart the pump shaft sheared. The pump was removed from the test stand and disassembled. The oil in the pump was blackened and contained many bronze particles as shown on the sleeve bearing in Figure 13. The bronze lands on the bottom of the piston shoes (Figure 14) were completely gone. The wear plate in Figure 15 was pitted, scarred, and blackened from excessive heating. Despite the many bronze particles, the remainder of the pump parts were apparently in good condition. There was cavitation erosion on the port plate (Figure 15) in the decompression area, but the silvered surface was intact.

A shoe failure results from the loss of the hydrodynamic pressure balance provided by a fluid film between the shoe and the wear plate. Oil for the shoe enters through a hole in the center of the piston shoe assembly. Should the hole become blocked, or the lands which control the thrust loads on the piston wear down, the piston shoe would either lose the lubricating oil or cause an uneven pressure distribution. Consequently the shoes would wipe against the wear plate. If pieces of the bronze land on the steel shoe were to dislodge, they would probably wedge between the shoes and wear plate creating excessive friction. (Some large pieces of bronze were found inside the pump case.)

Excessive wear of the shoe lands coupled with the high loads in the shoes caused by the pump running at maximum flow (100 hp), probably contributed to the uneven distribution and subsequent loss of the hydrodynamic pressure balance between a shoe and wear plate. Once one shoe started flaking particles, the others rapidly followed.

It is doubtful that any foreign objects entered the pump to cause the failure. The test system had 10μ nominal 25μ absolute filtration. The small particles from the cavitation erosion on the port plate would not be large enough to block the holes in the piston shoes. Excessive wear of the shoe lands at high load conditions probably caused an uneven pressure distribution between the shoes and wear plate that caused the shoes to wipe the plate and lock up the pump. A plating failure between the steel shoes and bronze lands is another though unlikely possibility.

The shoe and piston assembly had about 700 hrs of operating time. When the F-15 instrumented pump was refurnished only the barrel and port plate were replaced. It appears that after 4 years of testing, the lands finally gave out.

FIGURE 13 F-15 INSTRUMENTED PUMP SLEEVE BEARING

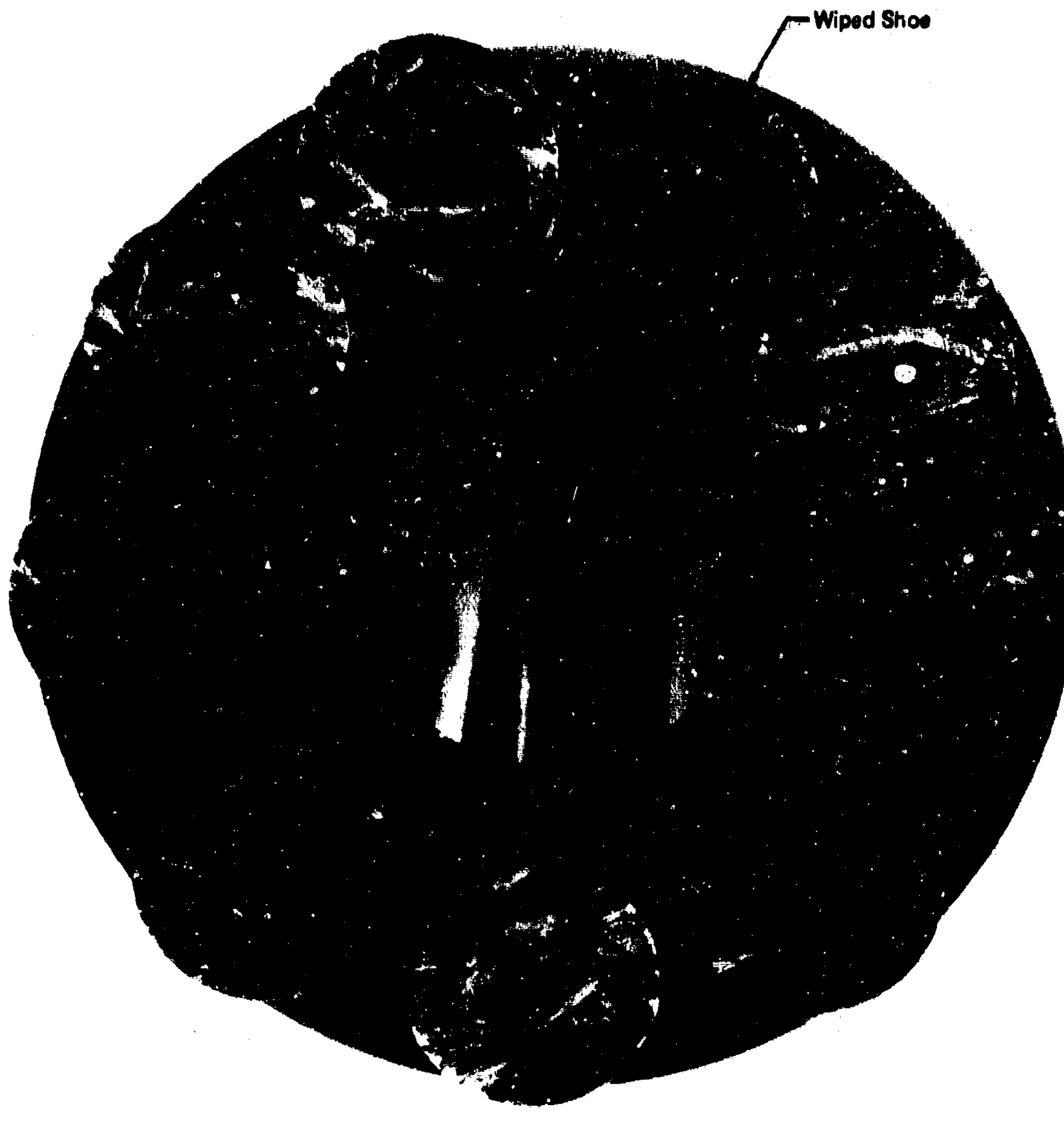

FIGURE 14 F-15 INSTRUMENTED PUMP PISTON SHOES

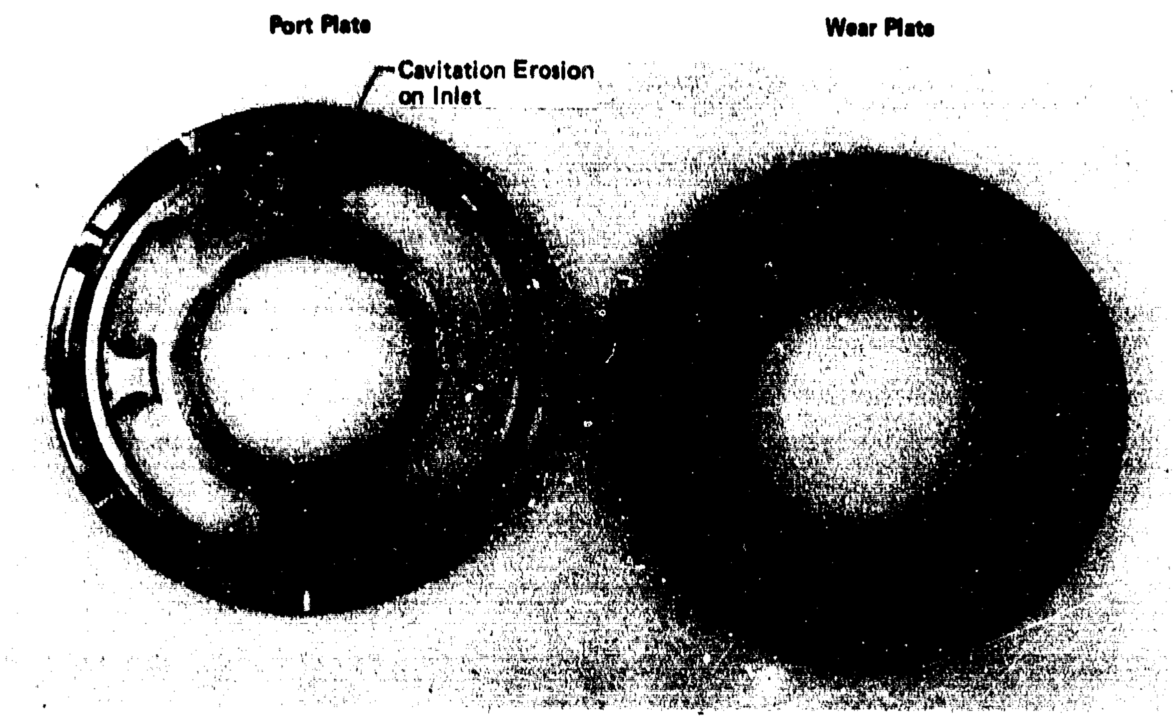

FIGURE 15 F-15 INSTRUMENTED PUMP PARTS

MCAIR was able to obtain a new hanger shoe and piston assembly from Abex. The unit arrived on 18 April 1979. After sufficient break-in time the tests were resumed. The steady state tests were completed and the 546 in^3 system was assembled. With the start of transient testing, problems were encountered with the hanger LVDT on the F-15 instrumented pump.

The LVDT core is attached to a rod which passes through a cylindrical barrel and rests on the pump hanger. The rod and barrel form a lap fitting between the case oil and atmosphere. The rod had worn with use creating a large leakage path from the pump case. The rod wear pattern was elliptical allowing it to wobble in the fitting. Whenever the hanger stroked, the rod stuck in the cylindrical barrel. The rod was removed and polished with an emory cloth several times, but it would still bind in the fully retracted position. The entire rod and barrel assembly needed to be reworked. This could not be accomplished within the allotted test period, so testing continued without the hanger LVDT.

Midway through the transient test series with the 546 in^3 system, the F-15 instrumented pump exhibited abnormally high torque values. The pump was removed from the HPAF and disassembled. A 1/8" dia bronze flake was found under the barrel. Several bronze particles and a steel sliver (3/32" long) were also discovered.

Contract budget limitations prohibited the rebuilding of the F-15 instrumented pump. MCAIR recommends that the entire unit be shipped to Abex for refurbishment.

An F-15 iron bird pump was used to finish the empirical pump test on the 546 in^3 system and 37 in^3 system.

An Abex F-4 pump (AP10V-62) was tested in 31, 270 and 538 in^3 systems without any problems.

HYTRAN empirical pump model verification tests were completed on 31 May 1979.

(1) Drive Torque Correlation with Hanger Position

When a sudden load change causes system pressure to vary from the regulated pump output pressure, the pump must adjust the output flow to meet the new load condition. The time elapsed during the pressure change is a measure of the dynamic response of the pump and system. The response depends on system compressibility, load resistance, pump speed and hanger rate.

The F-15 instrumented pump failed early in the empirical pump test program. The hanger position was lost. Fortunately the drive torque is able to provide a reasonable approximation to the hanger response.

An ideal hydraulic pump has a mechanical power output of

$$HP_{out} = T * N \tag{18}$$

Where

T = Drive Torque (in-lb)

N = Pump shaft speed (Rad/Sec)

The hydraulic power is defined as

$$HP_{in} = P * Q \tag{19}$$

Where

P = Pump Pressure Rise (PSI)

Q = Pump Outlet Flow (CIS)

The overall efficiency is defined as the ratio of actual horsepower output to the hydraulic power output.

$$\frac{HPout}{HPin} = \frac{T*N}{P*Q} = \eta_{OA} \tag{20}$$

Where η_{OA} = overall efficiency

The overall efficiency of most aircraft hydraulic pumps usually exceeds 85% at rated operating conditions. The overall efficiency is the product of the volumetric and mechanical efficiencies. The volumetric efficiency is the ratio of the ideal pump flow to the actual output flow,and the mechanical efficiency is the ratio of actual to ideal pump torque.

The steady state equation (20) shows the influence of pump output pressure and flow on drive torque. When a system load is applied, pressure, flow, torque, and overall efficiency will change. With an adequate drive system, pump speed will remain constant. Assuming that the efficiency will remain constant equation (20) can be solved for torque with pressure and flow the independent variables.

$$T = \eta_{OA} * \frac{P*Q}{N} \tag{21}$$

The torque value in equation (21) depends on the sensitivity of the pressure and flow variables. For a typical hydraulic transient, pump outlet flow will percentage wise vary more than the system pressure. For a turn-off transient in a 279 in^3 system, pump outlet pressure transiently reaches 3700 psi. The approximate transient flow is 30 CIS. The change in pressure was 22% above the steady state level. The change in flow was 38% of rated flow at the operating RPM. Consequently, the larger percentage change in flow will affect the torque calculation and therefore the shape of the torque curve. However, this does not apply whenever more system flow is being demanded than the pump can physically deliver. For this case, the hanger spring will hold the hanger at maximum flow until system pressure comes within the regualtion range of the compensator. Except for a delay caused by the outlet pressures acting on the compensator valve, which dumps actuator pressure to case, and allows the spring to stoke the hanger; the initial hanger response rate resembles pump outlet pressure.

The hanger position was measured during transient tests with the F-15 instrumented pump. As reported in References (1) and (2), a LVDT was used to track the hanger motion. The hanger position was able to show the damping characteristics of the pump/system and pump flow response to system flow demand.

A plot of drive torque versus time for a turn-off transient in a 235 in^3 system with 77.0 CIS steady state flow is shown in Figure 16. The recorded hanger position superimposed on the graph illustrates the phase relationship between the two measured signals.

The torque measurement by itself cannot yield transient pump outlet flow values. The torque exhibits pump frictional and inertia effects not part of the generated hydraulic power. The initial transient torque spike reaches 780 in-lbs in Figure 16. The hanger position is at 77.0 CIS pump outlet flow and the pump outlet pressure is 3550 psi. The pump flow would be over 100 CIS to generate the 780 in-lb torque. For a turn-off transient the flow will not be above steady state levels at the pump. Consequently the initial spike may be due to pump barrel rotational inertia.

The torque instrumentation relies on strain gages to provide torque values in a bandwidth from DC to 900 HZ. A 125 HZ wave is superimposed on the basic 16 HZ frequency on the torque signal in Figure 16. Much of this is attributable to pump frictional and inertia effects.

Figure 17 shows the relation between hanger position and pump outlet pressure in a 235 in^3 volume. There is a .004 sec delay between the rise of the pressure signal and the start of the transient on the hanger.

The hanger position and case pressure in Figure 18 indicate better phase correlation. The inlet pressure overplotted with the hanger position is shown in Figure 19 . The initial pressure transient response matches the exact time of the hanger response but subsequent inlet system dynamics interfer with the inlet pressure signal.

Overplots of hanger position on drive torque, pump outlet pressure, case pressure and pump inlet pressure are shown in Figures 20 through 23 for a turn-on transient in a 235 in^3 system. The hanger phasing shows best correlation with the drive torque. If the hanger position in Figure 21 were shifted so that the minimum position matched the pressure minimum, the pressure outlet responses would match the hanger response. This is not surprising since the pump must respond to the flow demanded by the system during this period.

In general the torque measurement can provide a reasonable time history of pump hanger response. Given a plot of hanger position versus time one can roughly estimate hanger response times from the torque curve. However, without direct hanger position as a comparison, the hanger rate can only be estimated if the pump should stroke between maximum and zero flow for a turn-off transient. This gives a beginning and ending point that can be converted to displacement with the proper hanger geometry. Turn-on transients at maximum flow conditions cannot track hanger position in the torque output because of the hanger spring contribution.

One must be cautioned against using these hanger rates as pure pump response. System size and fluid properties play an intricate role not easily separated from the pump dynamics. Pump flow response time can be defined based on a low volume system but the time will change as the system is modified. Test data with the F-15 pump in a 37.in^3 system shows the torque reaching a peak value for a turn-on transient in .02 sec, .034 seconds in a 279 in^3 system, and .052 sec in a 546 in^3 system. The steady state flow values were much less than the rated flow of the pump so the torque gives reasonable initial hanger response rise times.

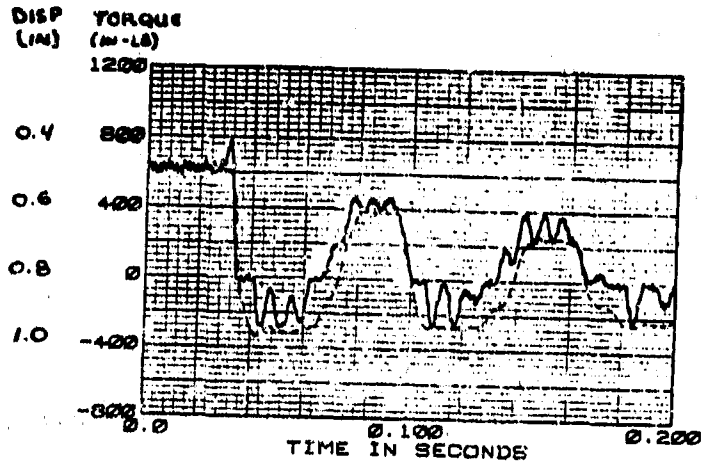

FIGURE 16 F-15 HYDRAULIC PUMP
84-A4-DT TURN-OFF TRANSIENT
77 CIS 130 F

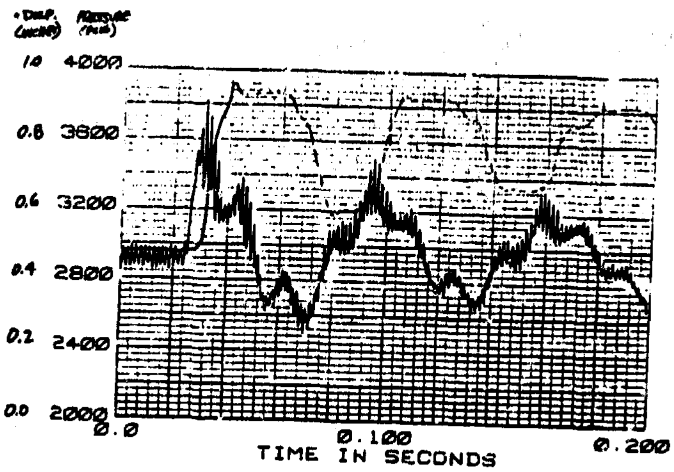

FIGURE 17 F-15 HYDRAULIC PUMP
84-A4-P3 TURN-OFF TRANSIENT
77 CIS 130 F

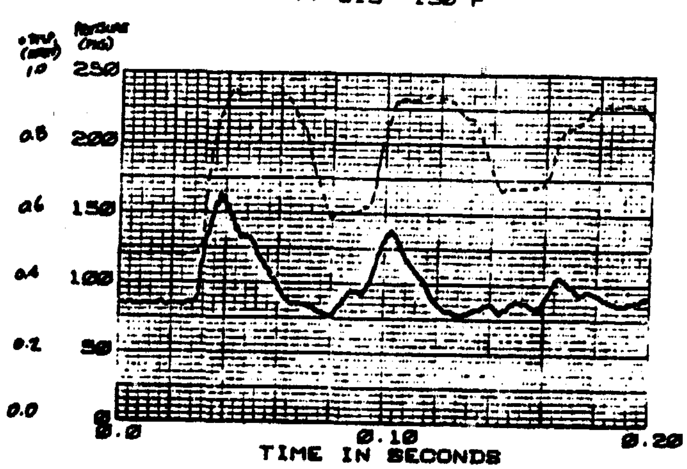

FIGURE 18 F-15 HYDRAULIC PUMP
84-A4-P1 TURN-OFF TRANSIENT
77 CIS 130 F

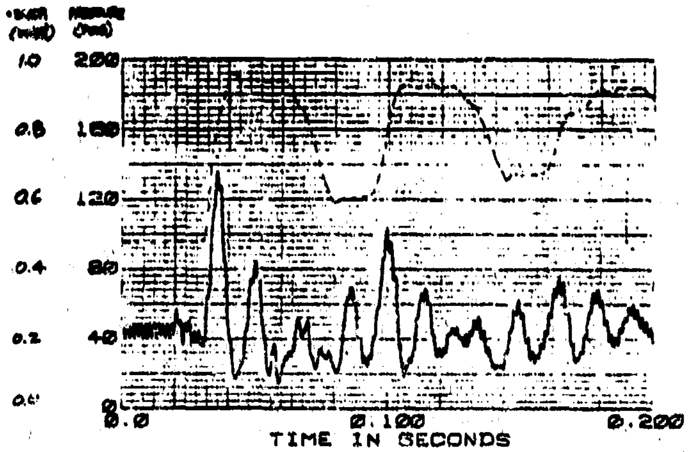

FIGURE 19 F-15 HYDRAULIC PUMP
94-A4-PS TURN-OFF TRANSIENT
77 CIS 130 F

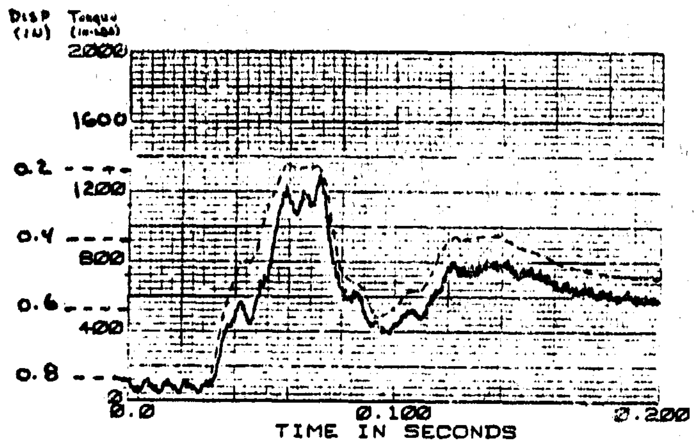

FIGURE 20 F-15 HYDRAULIC PUMP
94-A4+DT TURN-ON TRANSIENT
77 CIS 130 F

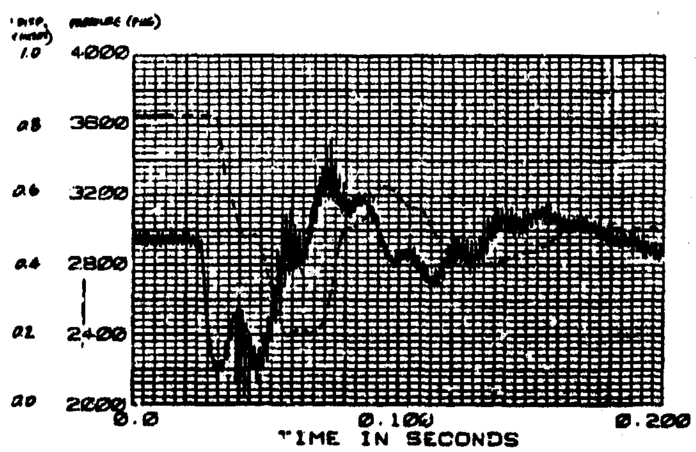

FIGURE 21 F-15 HYDRAULIC PUMP
94-A4+P3 TURN-ON TRANSIENT
77 CIS 130 F

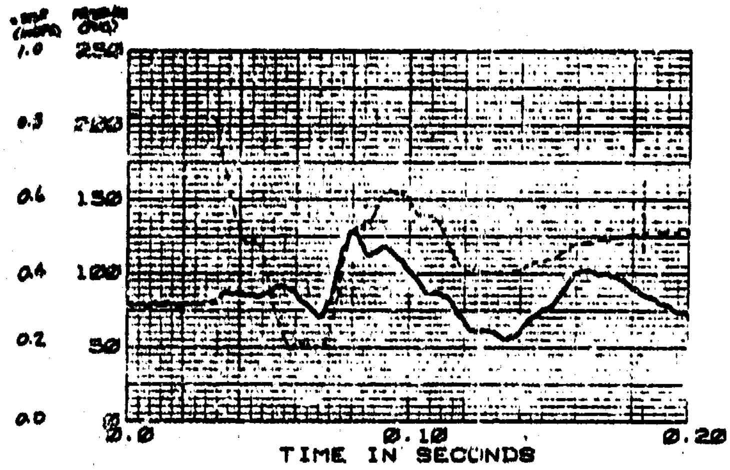

FIGURE 22 F-15 HYDRAULIC PUMP
S4-A4+P1 TURN-ON TRANSIENT
77 CIS 130 F

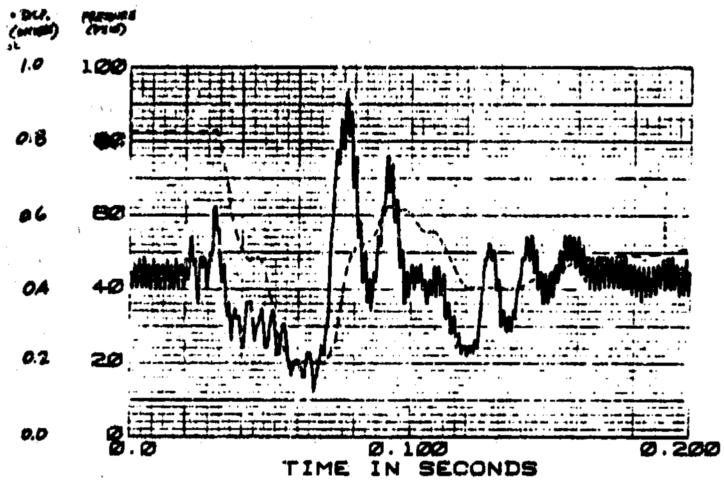

FIGURE 23 F-15 HYDRAULIC PUMP
S4-A4+PS TURN-ON TRANSIENT
77 CIS 130 F

For a high response pump, the drive torque or hanger position can provide a reasonable approximation to pump outlet flow. Generally the hanger position lags the actual flowrate. When a sudden flow step is applied, the compensator valve will take 20 to 50 msec to reach a new position and the system pressure will initially drop and recover within 2 to 5 msec because of the lead effect of the compensator valve. Figure 24 shows typical transient response curves of a pressure compensated pump and illustrates the relation between hanger position, flow, and pressure.

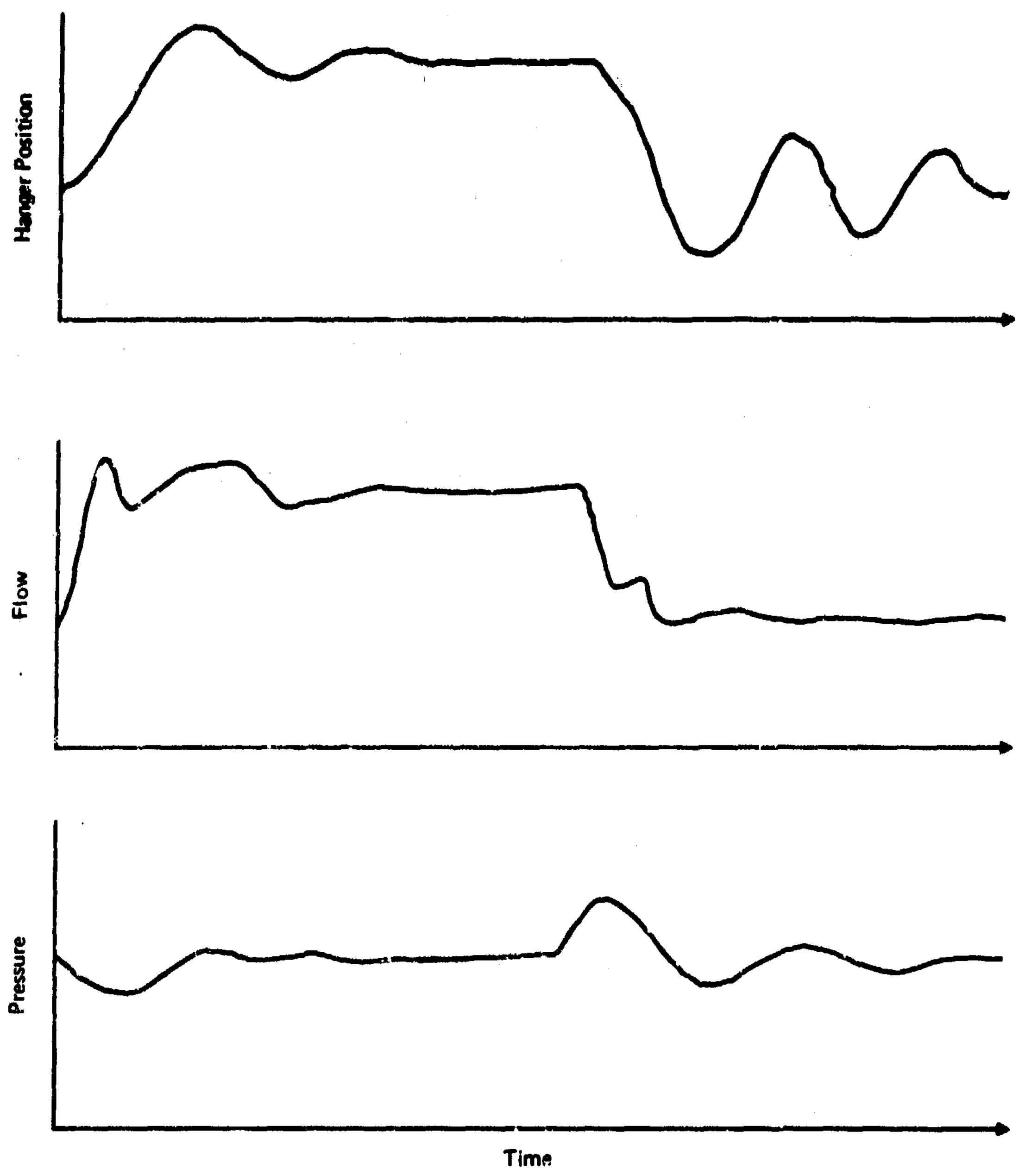

FIGURE 24 TYPICAL TRANSIENT RESPONSE OF PRESSURE COMPENSATED VARIABLE DISPLACEMENT PUMP

From previous pump testing (Reference 1), it was established that changes in the suction pressure at the pump inlet are mirrored in the actuator pressure that controls the hanger position. A graph of the actuator pressure for a turn-on transient is shown in Figure 25. The curve represents a rate of change in pump outlet flow, which can be graphically integrated to obtain pump outlet flow. The measured transient drive torque superimposed on the graph mirrors the outlet flowrate. Consequently pump outlet flow response can be approximated from the drive torque. The hanger position is also plotted on Figure 25 and illustrates the pumps high response rate. Figure 26 shows the same correlation for a turn-off transient.

(2) Steady State Testing

Pump performance is defined at the rated RPM which is located on every pump nameplate. Included with the rated RPM are the rated delivery at the maximum full flow pressure and the rated discharge pressure. The pump steady state performance can be determined by varying the load flow to obtain the controlled curve and then reducing the system back pressure to define the leakage curve.

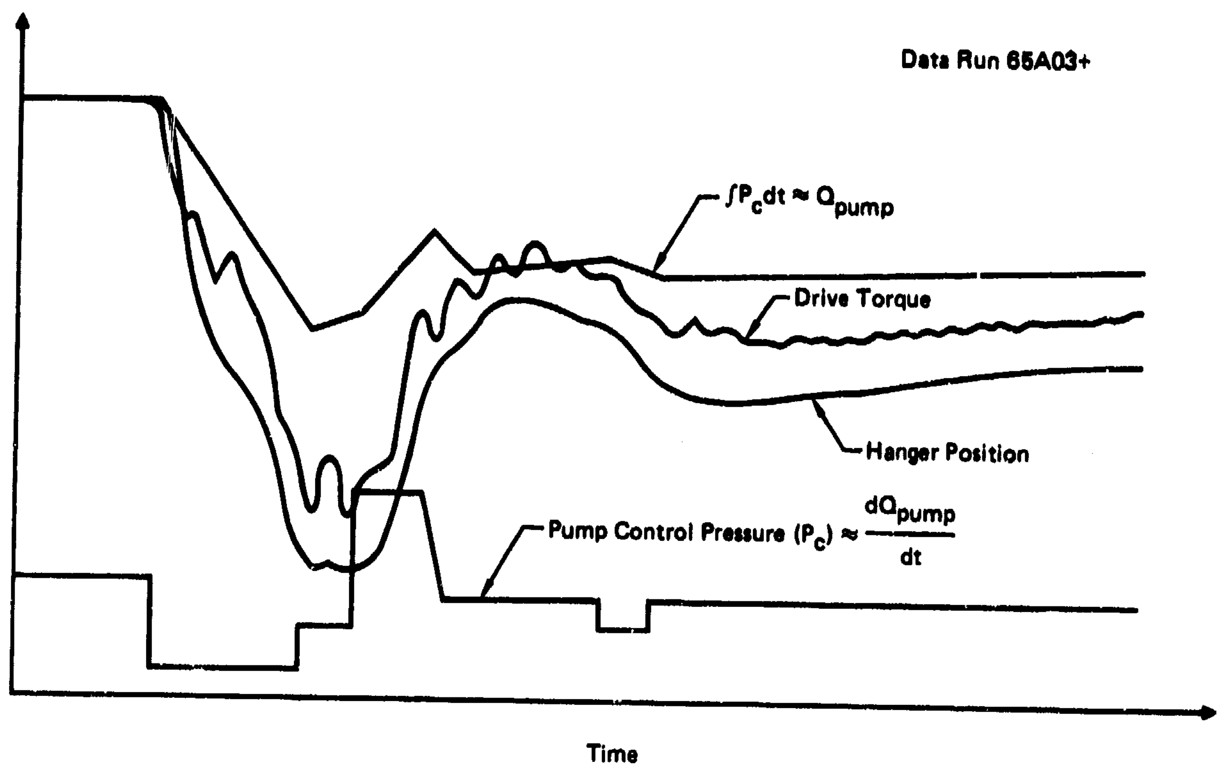

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FIGURE 25 RUN 65A03 TURN-ON TRANSIENT DATA COMPARISON

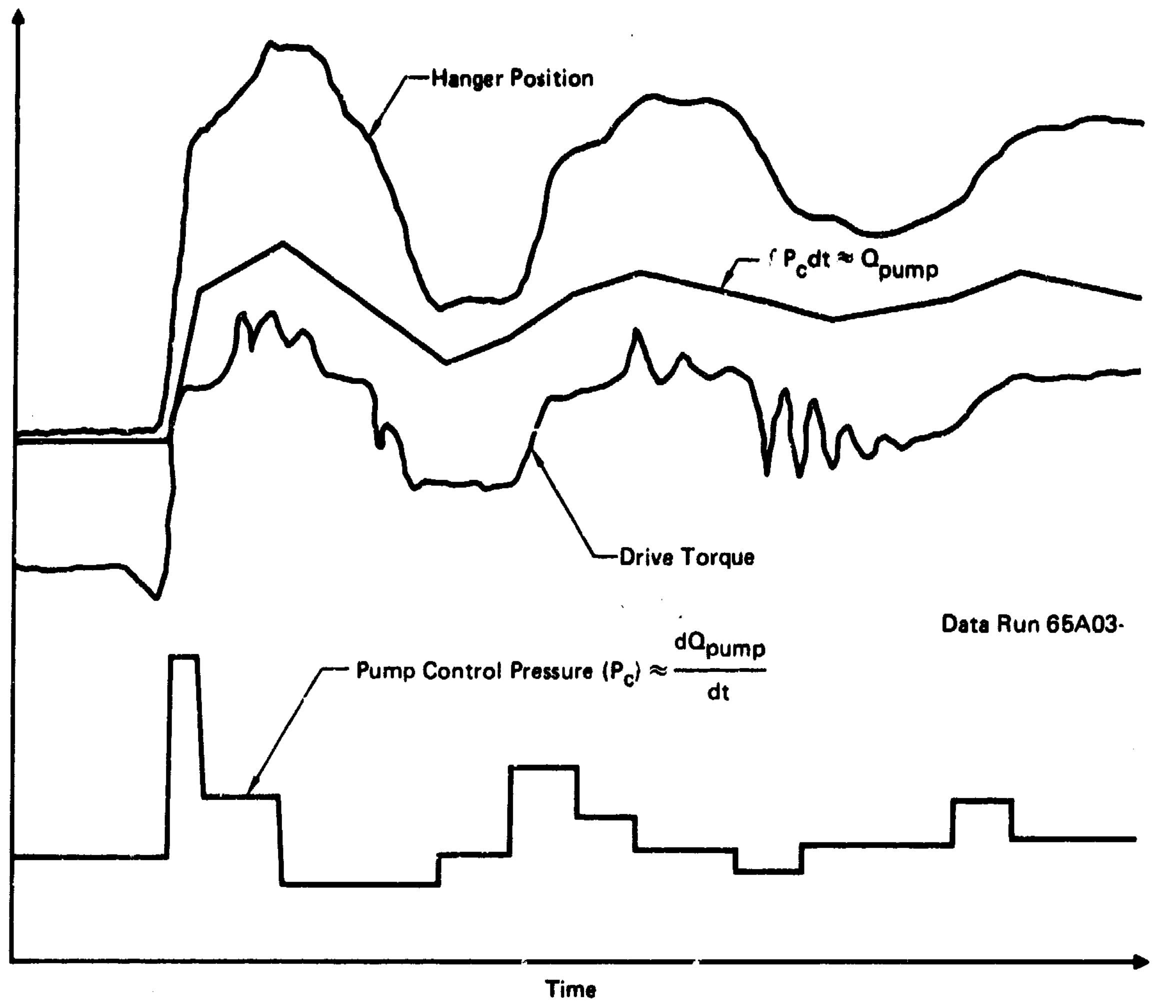


FIGURE 26 RUN 65A03 TURN-OFF TRANSIENT DATA COMPARISON

Steady state tests for the F-15 and F-4 pumps were run on the 278 in^3 and 270 in^3 volumes respectively. The transient control valve in Figure 9 was removed from the test stand for better control of system flow and back pressure with a load valve.

Test runs were made at 25, 50 and 100% pump rated RPM. Pump outlet temperature was 150°F and the F-4 bootstrap reservoir was set at 55 psig.

Plots of steady state outlet flow versus outlet pressure at 1100, 2300 and 4600 RPM for the F-15 hydraulic pump are shown in Figures 27, 28 and 29.

The case drain pressure versus flow plots in Figures 30, 31 and 32 were made at the pumps maximum compensated flow for 1100, 2300 and 4600 RPM. The plots were generated by recording both case flow and case pressure while closing a hand valve in the case drain line. The case drain plots are a reflection of the test system circuit effects on the leakage characteristics of the pump.

Plots of pump outlet flow versus pressure and case drain pressure versus flow for the F-4 pump are shown in Figures 33 through 38.

(3) Transient Testing

Pump dynamic performance cannot be separated from system dynamics. The response of the pump control system is not only determined by compensator characteristics but also by fluid bulk modulus and fluid volume under compression. These two factors will significantly alter the stability of the pump and its response to load pertubations. Consequently, three different volume systems were tested to measure pump/system response. An accumulator was added to the line system to measure the pump response to the extra inertia provided by the accumulator piston. A listing of the transient test runs with the F-15 and F-4 hydraulic pumps is presented in Table 2.

(a) F-15 Pump Transient Testing 37.27 in^3 System

Pump speed was limited to 1950 RPM because of compensator instability in the 37 in^3 system. For a 10 CIS turn-off transient the pump outlet pressure did oscillate after the load valve closed as shown in Figure 39. The pump outlet pressure for the turn-on transient was more stable (Figure 40).

At 50 CIS steady state flow the turn-off transient pump outlet pressure (Figure 41) does dampen out.

Recorded test data for turn-off and turn-on transients at 77 CIS steady state flow are shown in Figures 42 and 43. The pump inlet transducer (P2) was a Kistler Piezo-Electric Model 205 HZ. The transducer instrumentation outputs an electrical signal proportional to the oscillatory pressures. The transducer has the ability to record steady state values but because of the transducer sensitivity to temperature variations, these values will drift. Consequently the Y axis of the P2 trace in Figure 42 reflects the magnitude of the pressure pulsations.

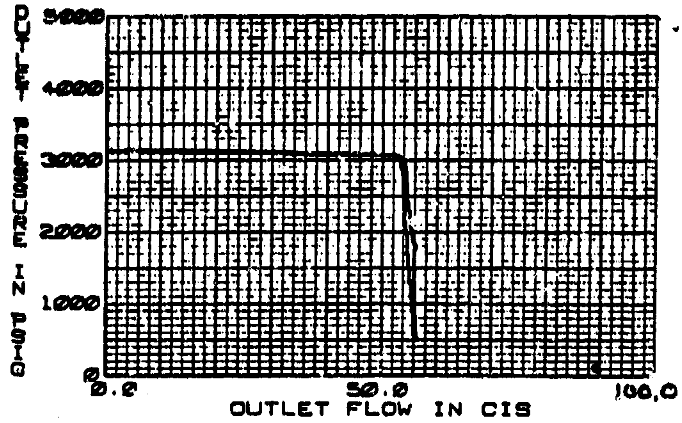


FIGURE 27 F-15 INST PUMP
STEADY STATE
1100 RPM 115 F

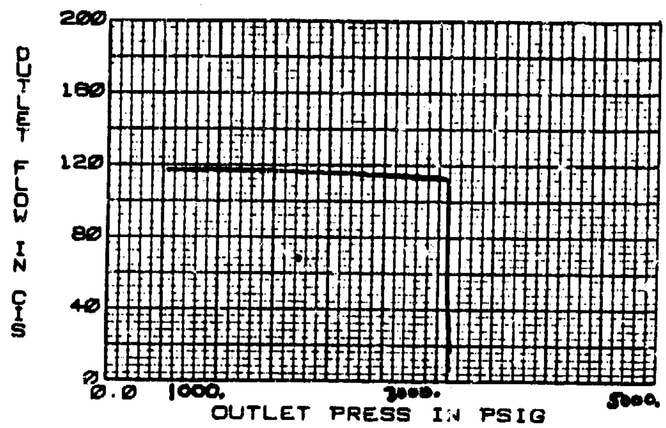


FIGURE 28 F-15 INST PUMP
STEADY STATE
2300 RPM 150 F

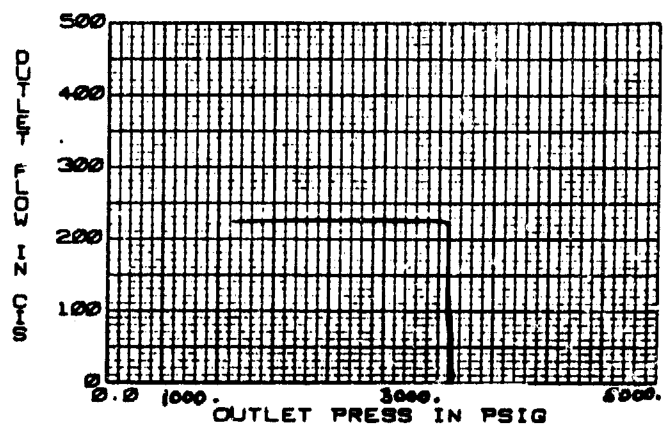


FIGURE 29 F-15 INST PUMP
STEADY STATE
4800 RPM 150 F

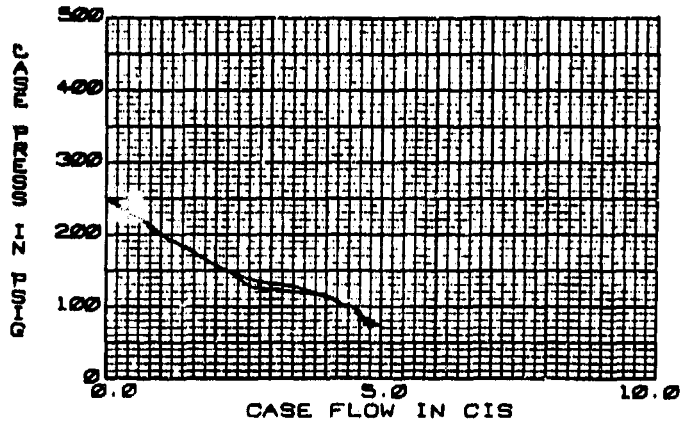


FIGURE 30 F-15 INST PUMP
1100 RPM
150 F
51 CIS MAIN FLOW

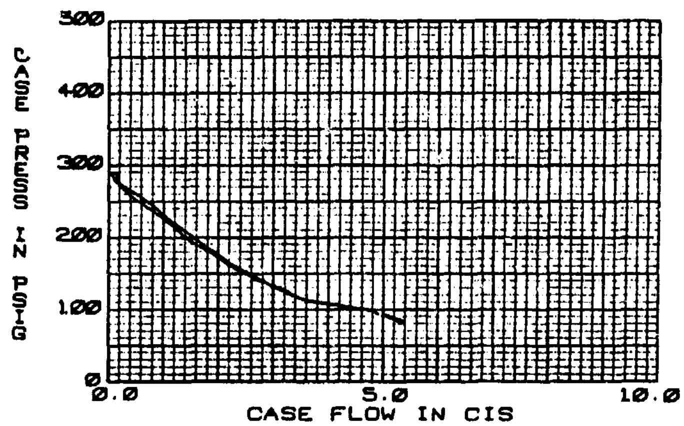


FIGURE 31 F-15 INST PUMP
2300 RPM
150 F
111 CIS MAIN FLOW

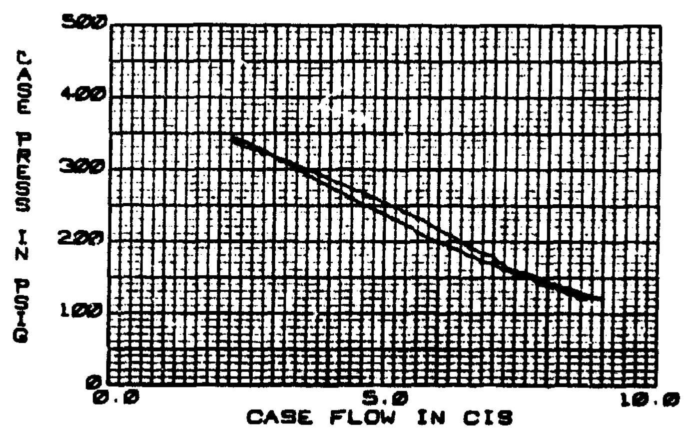


FIGURE 32 F-15 INST PUMP
4000 RPM
150 F
233 CIS MAIN FLOW

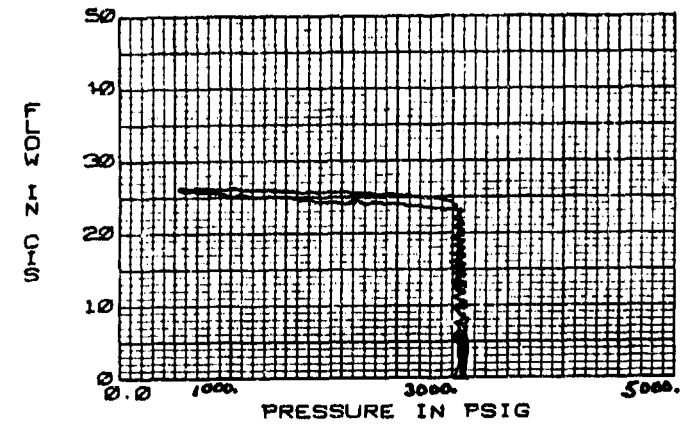


FIGURE 33 F-4 HYD PUMP
STEADY STATE
838 RPM 150 F

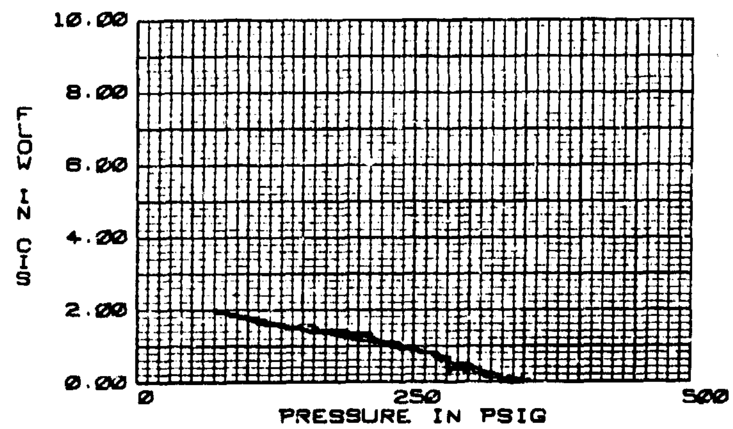


FIGURE 34 F-4 HYD PUMP
CASE DATA
838 RPM 150 F

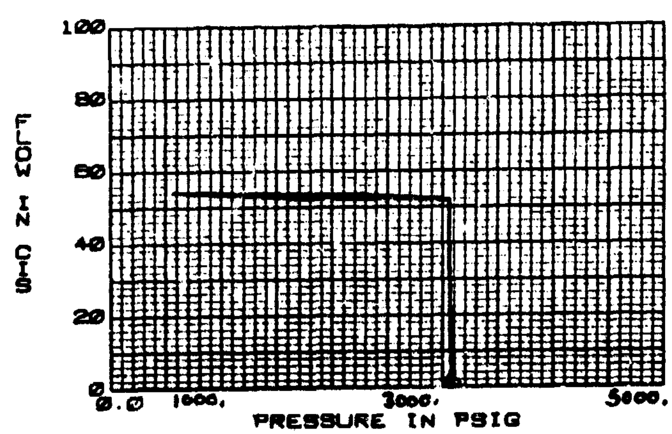


FIGURE 35 F-4 HYD PUMP
STEADY STATE
1875 RPM 157 F

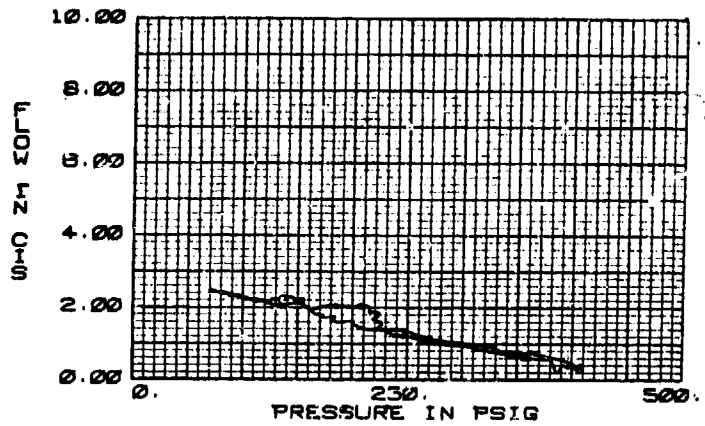


FIGURE 36 F-4 HYD PUMP
CASE DATA
1875 RPM 150 F

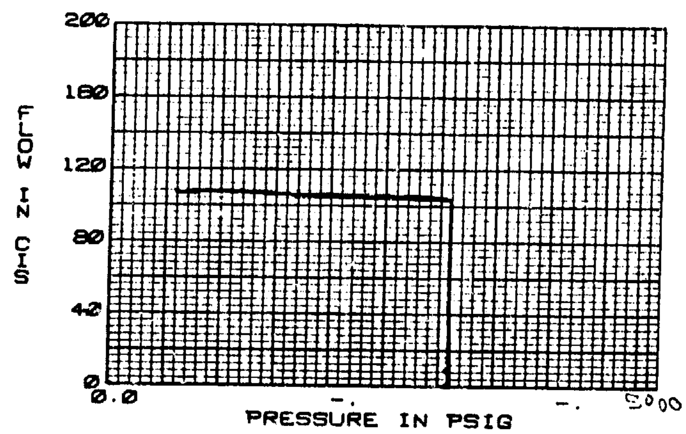


FIGURE 37 F-4 HYD PUMP
STEADY STATE
3750 RPM 150 F

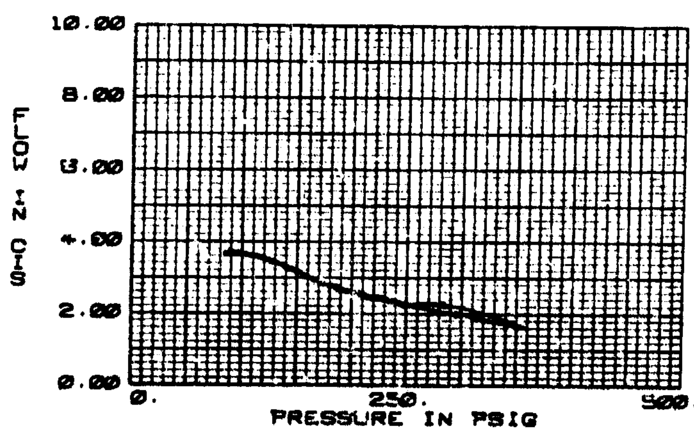


FIGURE 38 F-4 HYD PUMP
CASE DATA
3750 RPM 150 F

TABLE 2 HYTRAN EMPIRICAL PUMP MODEL TRANSIENT VERIFICATION TESTS

F-15 HYDRAULIC PUMP

STEADY STATE FLOW (CIS) LOW	HIGH	PUMP SPEED (RPM)	RESERVOIR PRESSURE (PSIG)	CASE PRESSURE (PSIG)	CASE FLOW (CIS)	PUMP OUTLET TEMP (°F)	SYSTEM VOLUME (IN^3)	RUN NUMBER
1.9	10.0	3750.	55.0	85.	4.47	149.	278.79	100-1-[1]
2.0	10.0	3750.	55.0	85.	4.62	154.	"	100-1+
2.0	50.0	3750.	53.0	85.	3.85	151	"	100-2-
2.0	50.0	3750.	57.0	83.	4.67	150	"	100-2+
2.0	77.0	3750.	50.0	85	3.40	150	"	100A3-
2.0	77.0	3750.	54.5	85	4.80	148	"	100A3+
2.0	107.0	3750.	55.0	85	4.01	155	"	100-4-
2.0	107.0	3750.	54.0	85	4.53	152	"	100-4+
2.0	77.0	3750.	53.5	83.	3.72	155	" [2]	100-5-
2.0	77.0	3750.	54.5	85.	4.85	155	" [2]	100-5+
2.0	108.0	3750.	54.5	86.	3.76	151	" [2]	100-6-
2.0	108.0	3750.	55.0	85.	4.86	154	" [2]	100-6+
2.0	50.0	3750.	56.0	82.	2.70	153	546.23	102-1-
2.0	50.0	3750.	50.0	85.	3.50	156	"	102-1+
2.0	100.0	3750.	56.0	77.	2.20	153	"	102-2-
2.0	100.0	3750.	50.0	85.	3.40	147	"	102-2+
2.0	117.0	3750.	57.0	77.	1.66	150	"	102-3-
2.0	117.0	3750.	51.0	85.	3.62	155	"	102-3+
2.0	117.0	3750.	55.0	77.	1.88	151	" [2]	102-4-
2.0	117.0	3750.	50.0	85.	3.58	148	" [2]	102-4+
2.0	100.0	3750.	52.0	75.	1.96	147	" [2]	102-5-
2.0	100.0	3750.	53.0	81.	3.31	150	" [2]	102-5+
2.0	50.0	3750.	51.0	84.	3.00	150	" [2]	102-6-
2.0	50.0	3750.	50.0	86.	3.39	143	" [2]	102-6+
2.0	10	1950.	57.	58.	2.31	93	37.27	103-1-
2.0	10	1950.	58.	58.	2.38	102	"	103-1+
2.0	50	1950.	58.	61.	1.38	110	"	103-2-
2.0	50	1950.	54.	56.	2.54	115	"	103-2+
2.0	77	1950.	55.	61.	1.38	112	"	103-3-
2.0	77	1950.	54.	56.	2.54	108	"	103-3+
2.0	95	1950.	55.	65.	1.65	110	"	103-4-
2.0	95	1950.	54.	55.	2.65	112	"	103-4+
2.0	95	1950.	54.	63.	1.54	109	" [2]	103-5-
2.0	95	1950.	50.	50.	2.50	110	" [2]	103-5+
2.0	77	1950.	53.	60.	1.27	105	" [2]	103-6-
2.0	77	1950.	50.	50.	2.46	110	" [2]	103-6+
2.0	50	1950.	50.	54.	1.34	100	" [2]	103-7-
2.0	50	1950.	50.	50.	2.38	102	" [2]	103-7+

F-4 HYDRAULIC PUMP

STEADY STATE FLOW (CIS) LOW	HIGH	PUMP SPEED (RPM)	RESERVOIR PRESSURE (PSIG)	CASE PRESSURE (PSIG)	CASE FLOW (CIS)	PUMP OUTLET TEMP (°F)	SYSTEM VOLUME (IN^3)	RUN NUMBER
2.0	10	3750.	58.	61.	3.20	108	31.3	104-1-
2.0	10	3750.	54.	56.	3.27	111	"	104-1+
2.0	50	3750.	58.	63.	3.04	113	"	104-2-
2.0	50	3750.	54.	56.	3.38	107	"	104-2+
2.0	77	3750.	58.	67.	2.92	110	"	104-3-
2.0	77	3750.	54.	56.	3.46	109	"	104-3+
2.0	104	3750.	59.	72.	2.77	110	"	104-4-
2.0	104	3750.	55.	56.	3.38	107	"	104-4+
2.0	104	3750.	55.	69.	3.85	104	" [2]	104-5-
2.0	104	3750.	50.	54.	4.15	106	" [2]	104-5+
2.0	77	3750.	55.	66.	3.65	108	" [2]	104-6-
2.0	77	3750.	50.	53.	3.73	108	" [2]	104-6+

1 - DENOTES TURN-OFF TRANSIENT
\+ DENOTES TURN-ON TRANSIENT

2 RUNS WITH JFS ACCUMULATOR IN THE SYSTEM

TABLE 2 (CONT.) HYTRAN EMPIRICAL PUMP MODEL VERIFICATION TESTS

F-4 HYDRAULIC PUMP

STEADY STATE FLOW (CIS) LOW	HIGH	PUMP SPEED (RPM)	RESERVOIR PRESSURE (PSIG)	CASE PRESSURE (PSIG)	CASE FLOW (CIS)	PUMP OUTLET TEMP (°F)	SYSTEM VOLUME (IN^3)	RUN NUMBER
2.0	66	3750.	51.	57.	2.38	98	31.3 [2]	104-7-
2.0	66	3750.	51.	52.	2.96	108	" [2]	104-7+
2.0	50	3750.	52.	56.	2.69	98	" [2]	104-8-
2.0	50	3750.	51.	52.	2.96	107	" [2]	104-8+
2.0	10	3750.	53.	56.	4.07	149	538.1	105-1-
2.0	10	3750.	53.	56.	3.94	156	"	105-1+
2.0	50	3750.	58.	61.	3.21	159	"	105-2-
2.0	50	3750.	53.	56.	3.77	161	"	105-2+
2.0	77	3750.	58.	64.	3.42	151	"	105-3-
2.0	77	3750.	53.	50.	3.82	149	"	105-3+
2.0	104	3750.	58.	64.	3.42	145	"	105-4-
2.0	104	3750.	53.	56.	4.02	148	"	105-4+
2.0	104	3750.	50.	61.	3.50	146	" [2]	105-5-
2.0	104	3750.	49.	50.	3.62	144	" [2]	105-5+
2.0	77	3750.	53.	61.	3.68	147	" [2]	105-6-
2.0	77	3750.	50.	51.	3.78	145	" [2]	105-6+
2.0	10	3750.	55.	55.	3.86	157	270.66	106-1-
2.0	10	3750.	54.	56.	3.64	158	"	106-1+
2.0	50	3750.	59.	69.	3.20	150	"	106-2-
2.0	50	3750.	55.	56.	3.62	150	"	106-2+
2.0	77	3750.	55.	61.	3.60	156	"	106-3-
2.0	77	3750.	54.	56.	3.62	155	"	106-3+
2.0	104	3750.	54.	66.	3.94	148	"	106-4-
2.0	104	3750.	54.	56.	3.78	153	"	106-4+
2.0	104	3750.	52.	63.	4.14	153	" [2]	106-5-
2.0	104	3750.	51.	52.	3.98	153	" [2]	106-5+
2.0	77	3750.	55.	64.	4.10	146	" [2]	106-6-
2.0	77	3750.	50.	53.	4.18	151	" [2]	106-6+

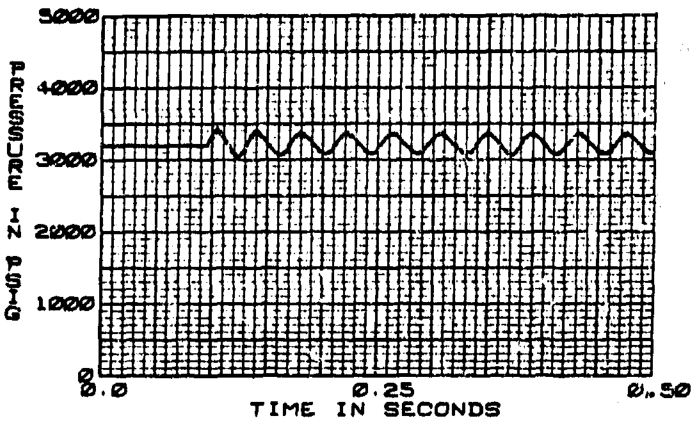


FIGURE 39 F-15 HYD PUMP
103-1-P1 TURN-OFF TRANSIENT
10 CIS 100 F

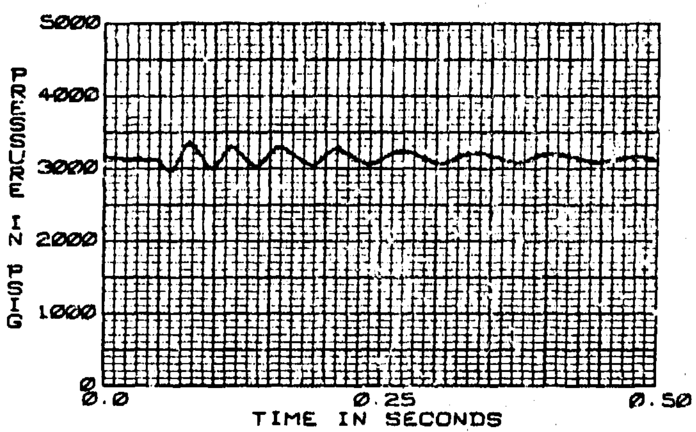


FIGURE 40 F-15 HYD PUMP
103-1+P1 TURN-ON TRANSIENT
10 CIS 100 F

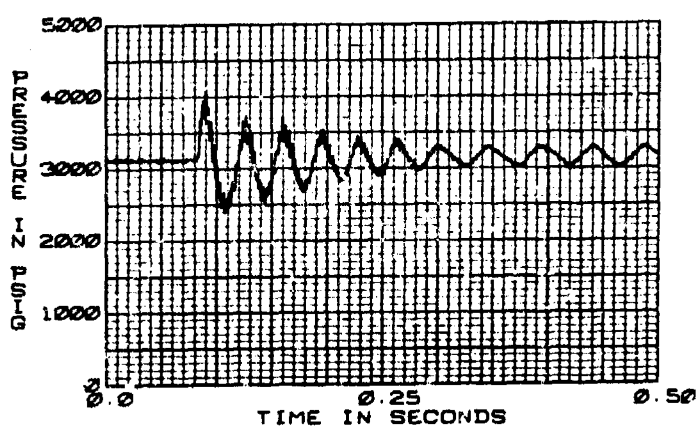


FIGURE 41 F-15 HYD PUMP
103-2-P1 TURN-OFF TRANSIENT
50 CIS 100 F

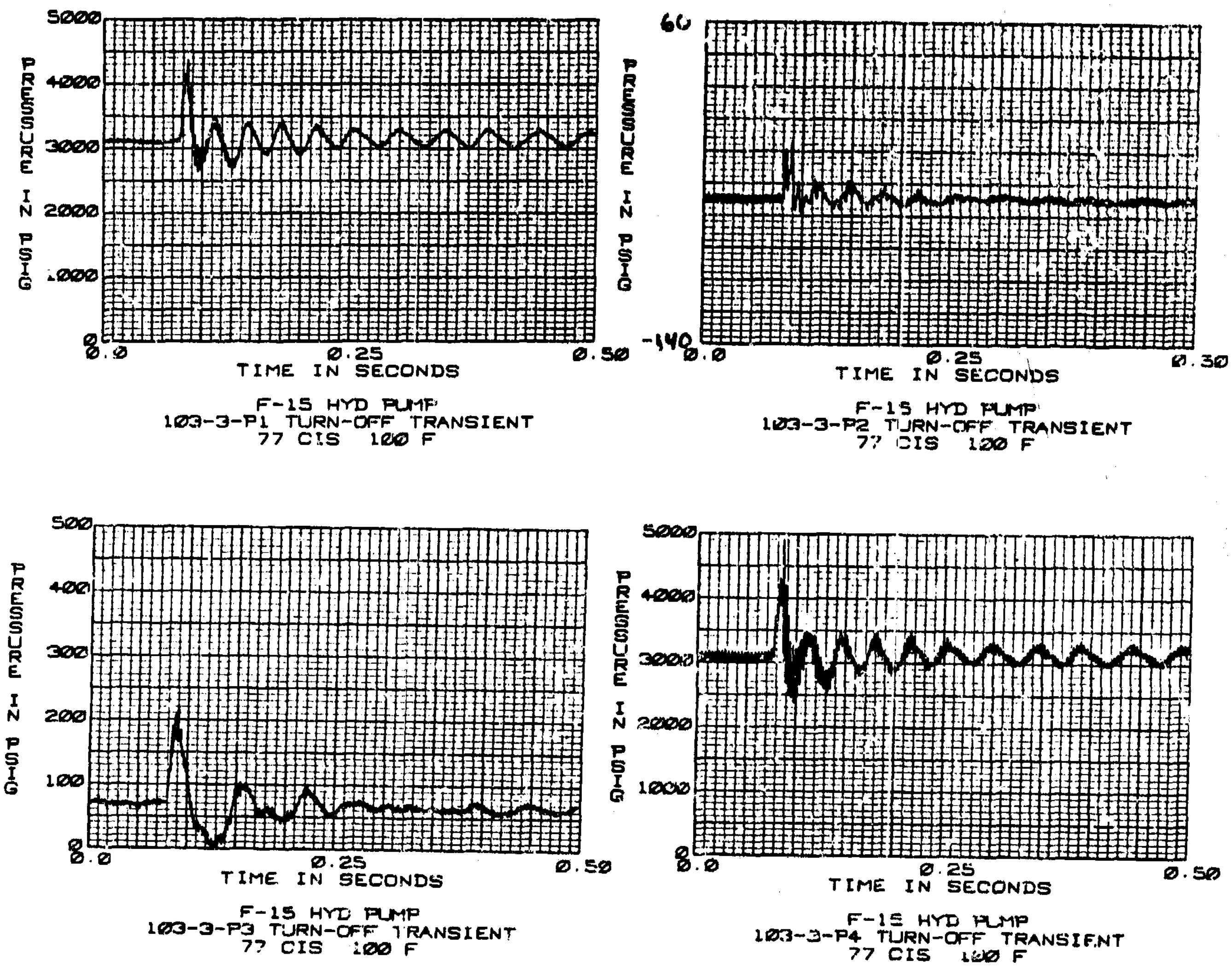
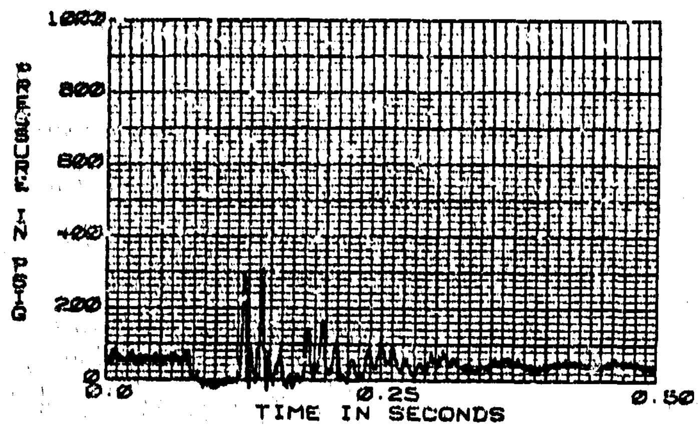
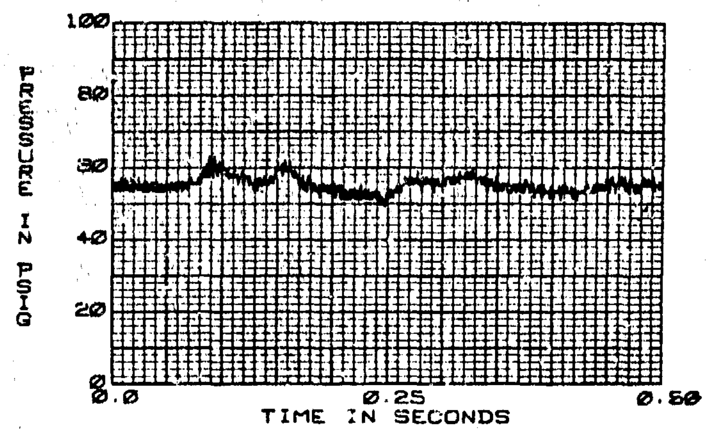


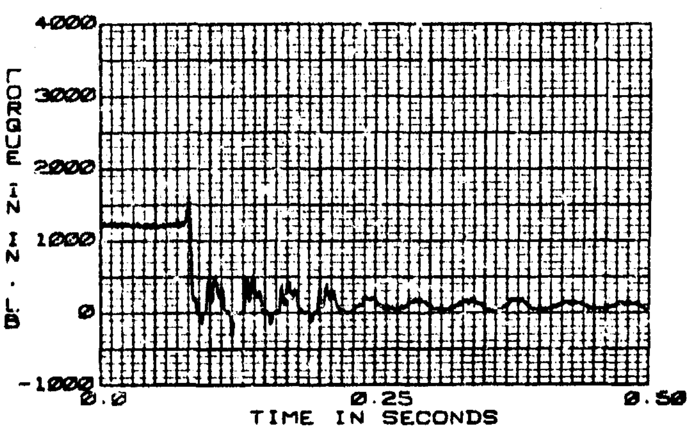
FIGURE 42 F-15 HYDRAULIC PUMP 103-3 TURN-OFF TRANSIENT, 77 CIS, 100°F



F-15 HYD PUMP
103-3-PS TURN-OFF TRANSIENT
77 CIS 100 F

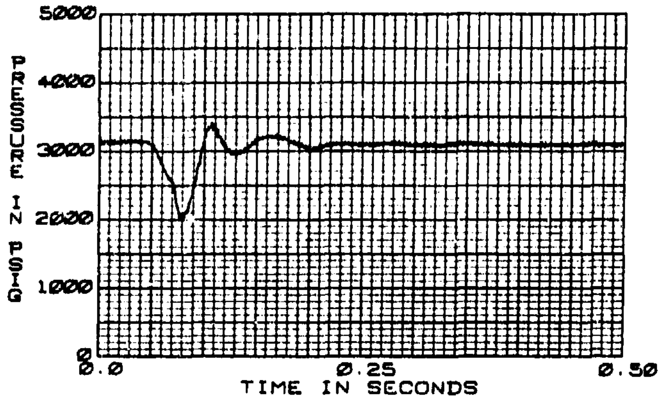


F-15 HYD PUMP
103-3-P7 TURN-OFF TRANSIENT
77 CIS 100 F

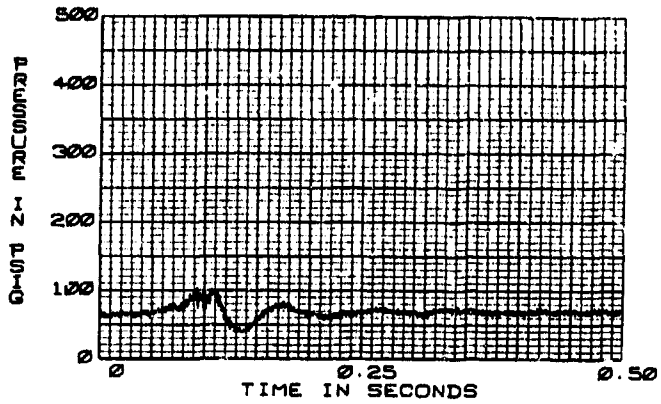


F-15 HYD PUMP
103-3-DT TURN-OFF TRANSIENT
77 CIS 100 F

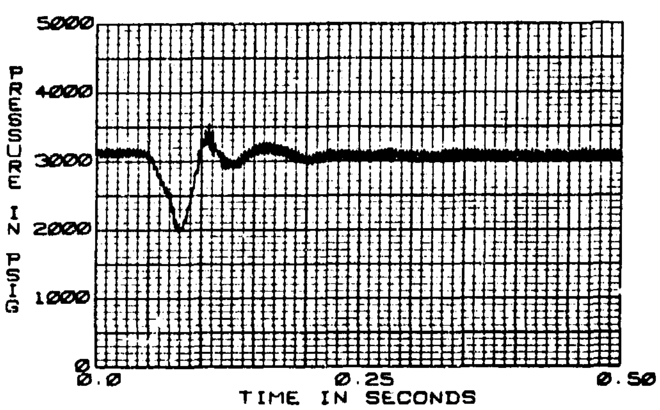
FIGURE 42 (CONTINUED)



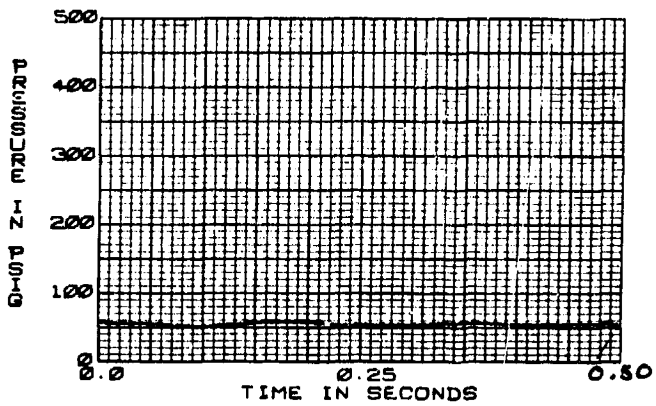
F-15 HYD PUMP
103-3+P1 TURN-ON TRANSIENT
77 CIS 100 F



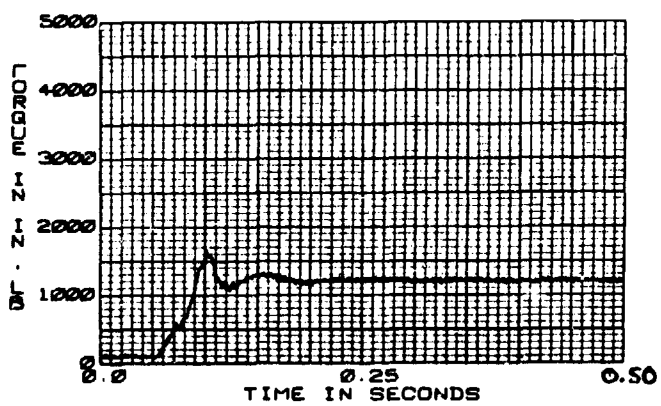
F-15 HYD PUMP
103-3+P3 TURN-ON TRANSIENT
77 CIS 100 F



F-15 HYD PUMP
103-3+P4 TURN-ON TRANSIENT
77 CIS 100 F



F-15 HYD PUMP
103-3+P7 TURN-ON TRANSIENT
77 CIS 100 F



F-15 HYD PUMP
103-3+DT TURN-ON TRANSIENT
77 CIS 100 F

FIGURE 43 F-15 HYDRAULIC PUMP 103-3 TURN-ON TRANSIENT, 77 CIS, 100°F

The steady state value was close to the reservoir pressure (P7). To obtain an inlet pressure, take the steady state pressure level then add or subtract the pulsation magnitude of P2 depending on the direction of the pressure spike.

The pump outlet pressure for the turn-on transient in Figure 43 dropped to 2000 psi in the 37 in^3 system. The pump was operating at 42% of its rated RPM.

When the JFS Accumulator was added to the 37 in^3 system, the pump outlet pressure did not fall below 2900 psi (Figure 44) for a turn-on transient. Accumulator oil and gas pressure and piston position are presented in Figure 45, for a turn-off transient at 77 CIS pump outlet flow. The pump outlet pressure P1 did not rise above 3450 psi.

The JFS Accumulator has a 140 in^3 maximum oil volume and 70 in^3 min gas volume. The piston area on the oil side is 18.892 in^2 with an 8 inch total stroke. For the turn-on transient the piston moved 0.1" and displaced 1.88 in^3 of oil. The absence of a large load on the test stand prohibited a good transient study of the accumulator in the 37 in^3 system.

(b) F-15 Pump Transient Testing 278.79 in^3 System

Turn-on and turn-off transients were run for 10, 50, 77 and 107.8 CIS steady state pump outlet flows. The pump outlet (P1) case drain (P3) and inlet (P2) pressures for a 77. CIS turn-off transient are shown in Figure 46. The hanger position (XH) was obtained before the F-15 instrumented pump failed. The transient pressure (P4) trace next to the load valve is shown in Figure 46. The P1, P4 and XH parameters for the turn-on transient at 77 CIS are shown in Figure 47.

The accumulator was added to the system and runs were made at 77. and 107.8 CIS steady state pump flows. The results for the turn-off and turn-on transients at 77 CIS are presented in Figures 48 and 49. The pump outlet pressure dropped to 2500 psi for the turn-on transient without the accumulator. With the accumulator in the circuit 2900 psi was the minimum pressure.

(c) F-15 Pump Transient Testing 546.23 in^3 System

Turn-off and turn-on transients were run in the 546 in^3 system for 50, 100 and 117 CIS steady state flow rates. Transient data for the turn-off and turn-on runs are presented in Figures 50 and 51. Figures 52 and 53 present the transient test results when the JFS accumulator was added to the test circuit.

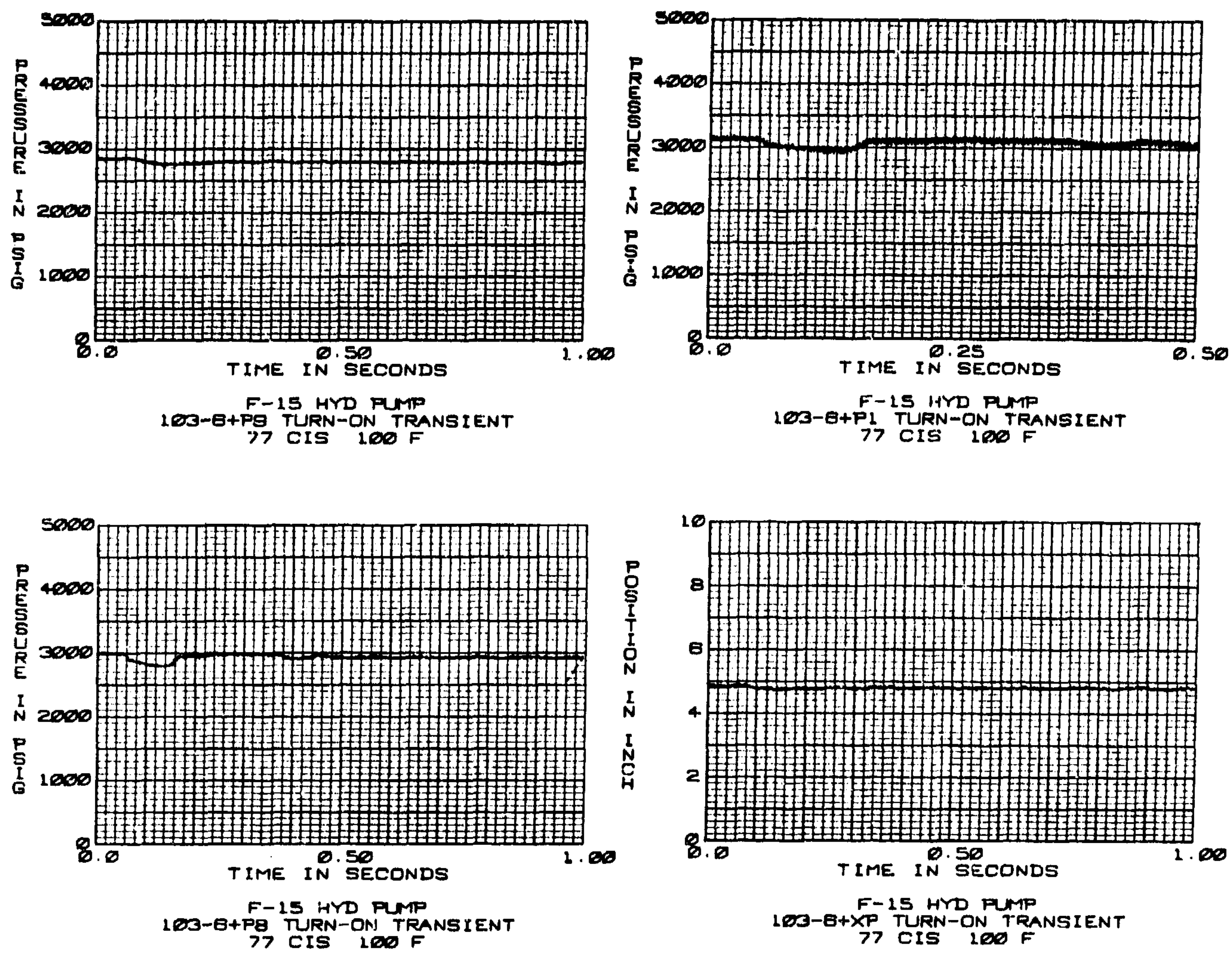


FIGURE 44 F-15 HYDRAULIC PUMP 103-6 TURN-ON TRANSIENT, 77 CIS, 100°F

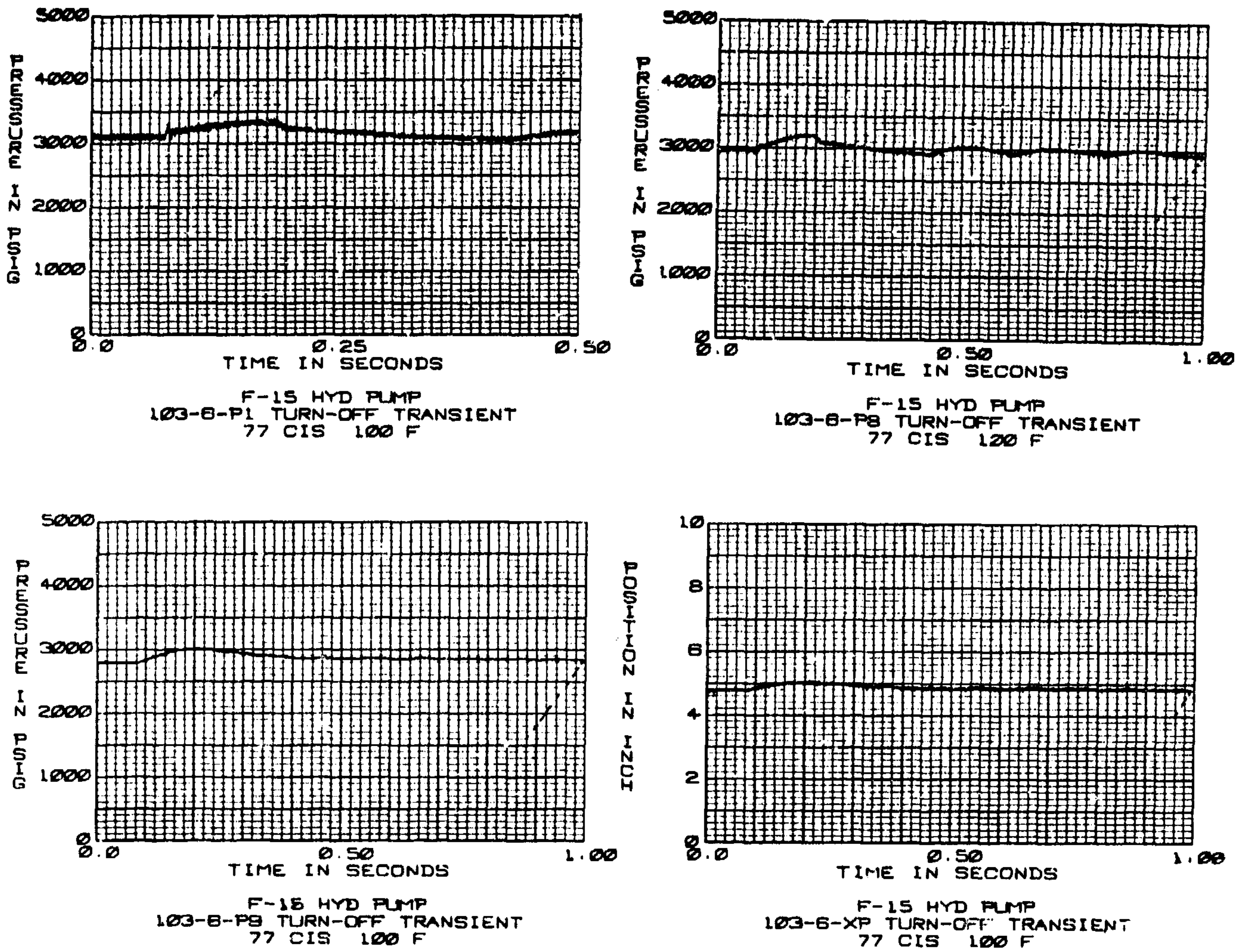


FIGURE 45 F-15 HYDRAULIC PUMP 103-6 TURN-OFF TRANSIENT, 77 CIS, 100°F

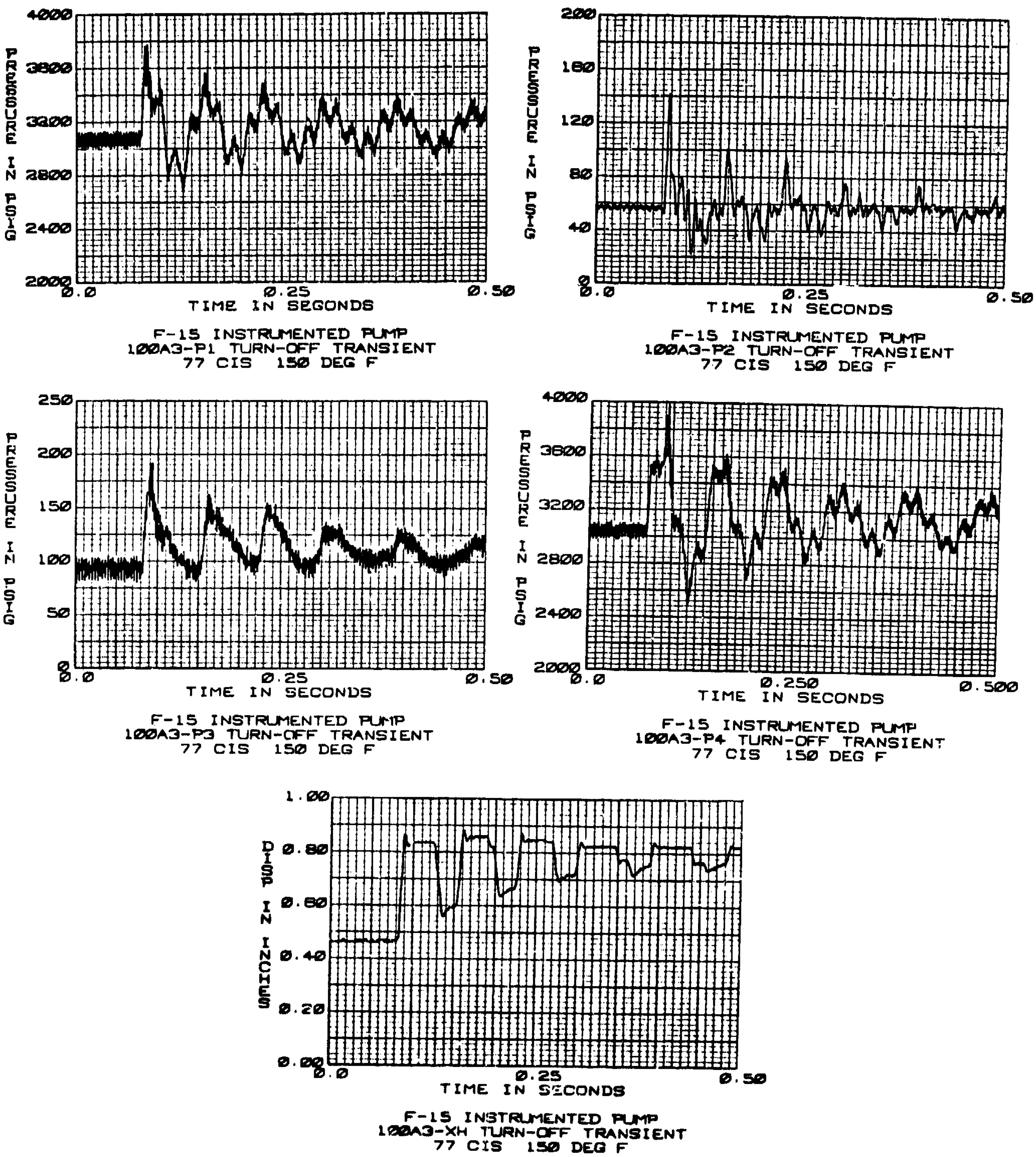
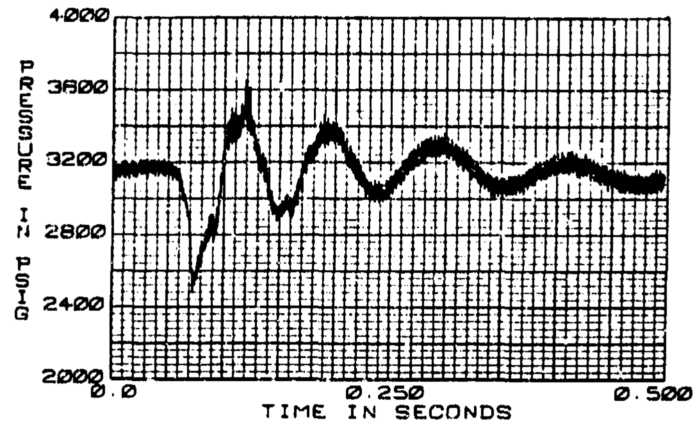
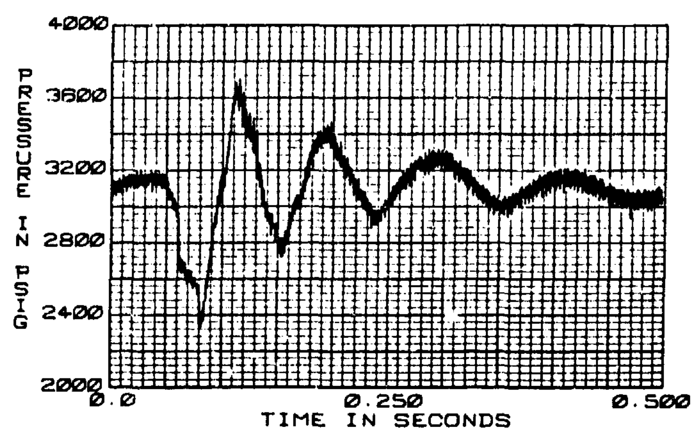


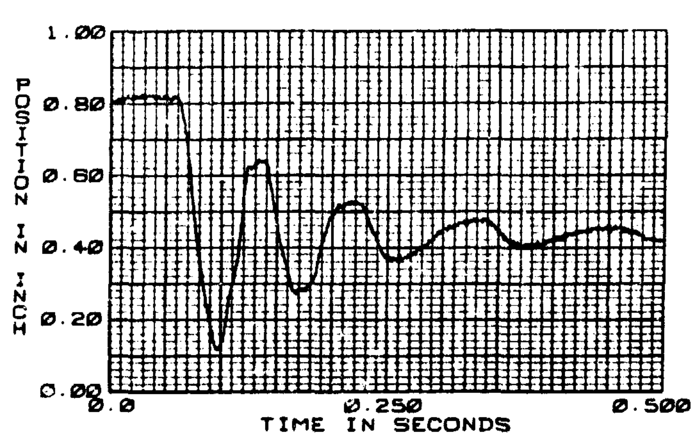
FIGURE 46 F-15 HYDRAULIC PUMP 100A3 TURN-OFF TRANSIENT, 77 CIS, 150°F



HYTRAN--F-15
100A3+P1 TURN-ON TRANSIENT
20 GPM 150 F

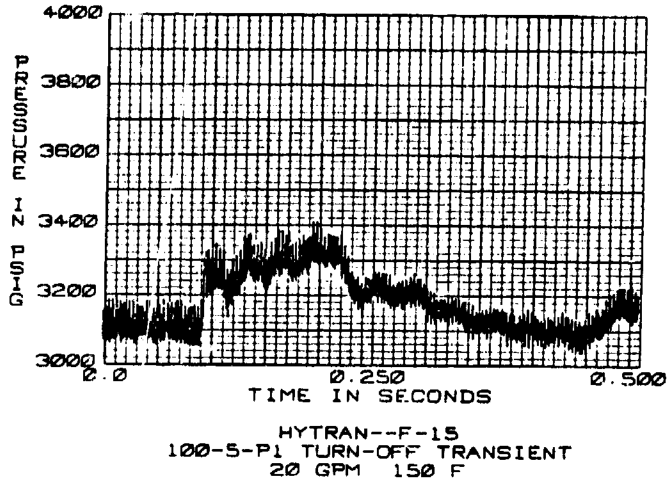


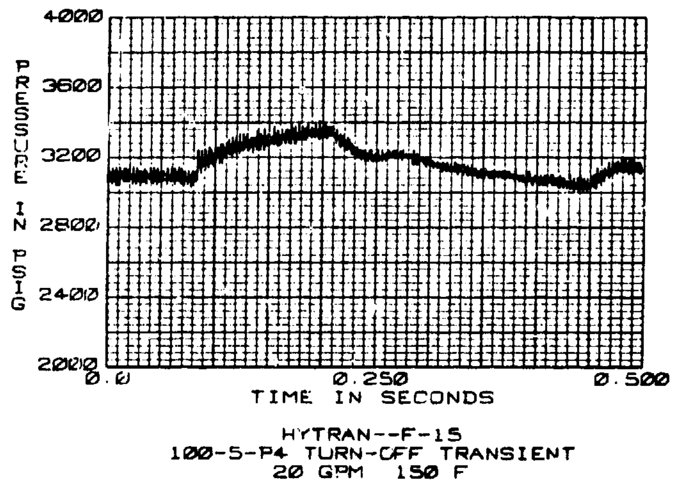
HYTRAN--F-15
100A3+P4 TURN-ON TRANSIENT
20 GPM 150 F



HYTRAN--F-15
100A3+XH TURN-ON TRANSIENT
20 GPM 150 F

FIGURE 47 F-15 HYDRAULIC PUMP 100A3 TURN-ON TRANSIENT, 77 CIS, 150°F





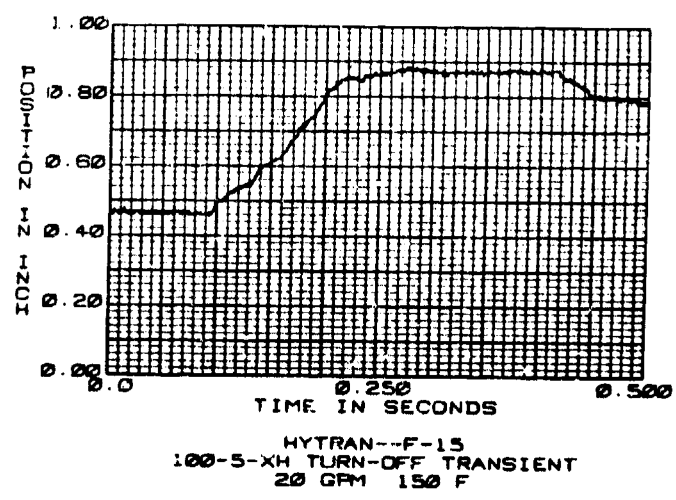
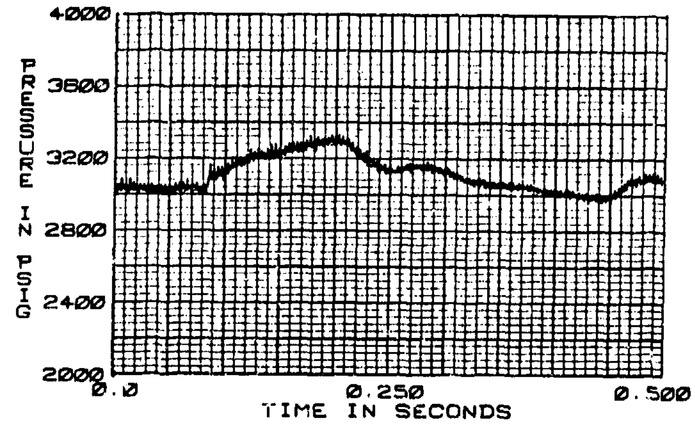
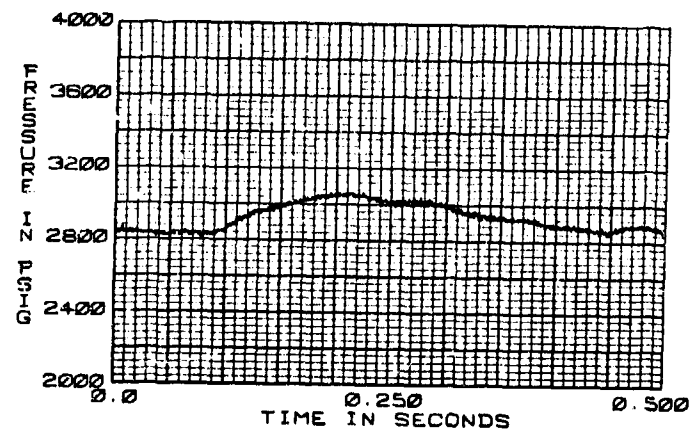


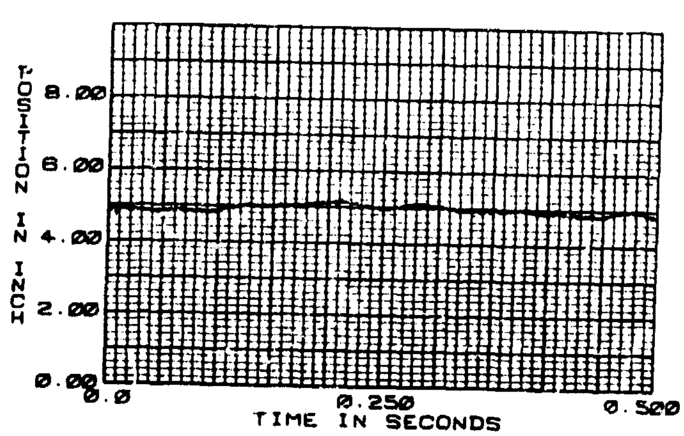
FIGURE 48 F-15 HYDRAULIC PUMP 100-5 TURN-OFF TRANSIENT, 77 CIS, 150°F



HYTRAN--F-15
100-5-P8 TURN-OFF TRANSIENT
20 GPM 150 F

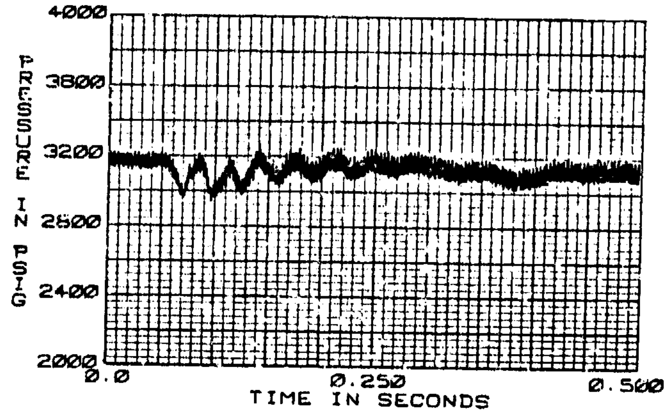


HYTRAN--F-15
100-5-P8 TURN-OFF TRANSIENT
20 GPM 150 F

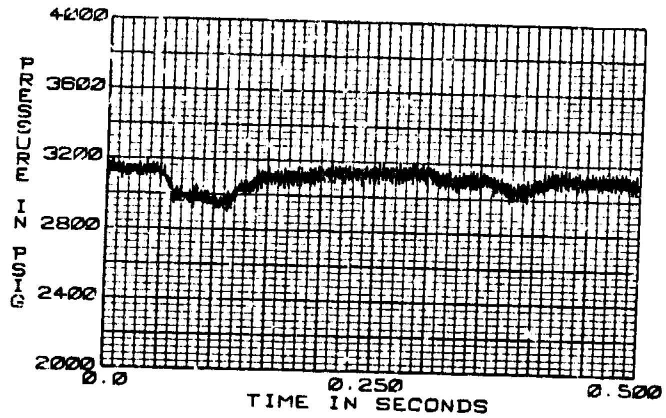


HYTRAN--F-15
100-5-XP TURN-OFF TRANSIENT
20 GPM 150 F

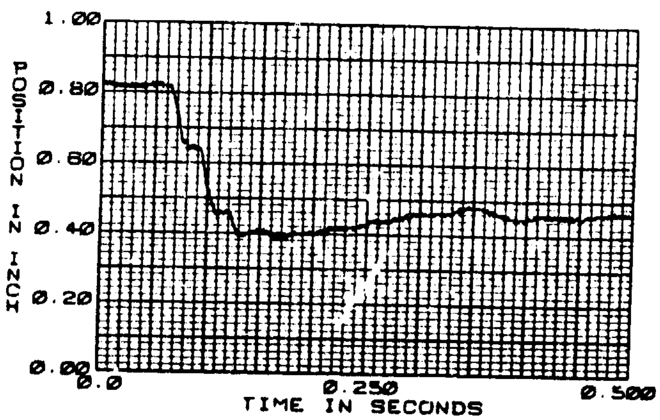
FIGURE 48 (CONTINUED)



HYTRAN--F-15
100-5+P1 TURN-ON TRANSIENT
20 GPM 150 F

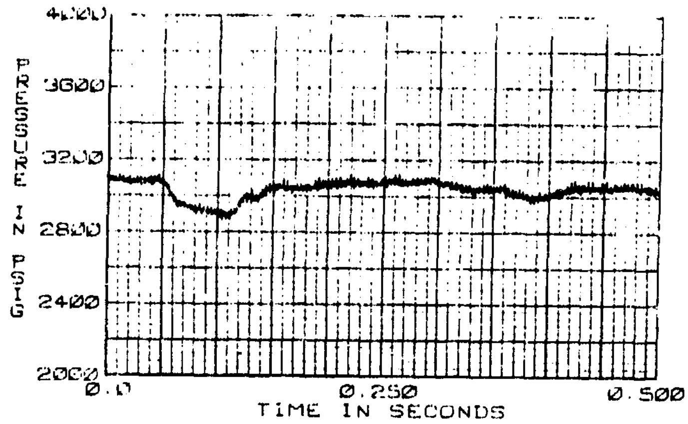


HYTRAN--F-15
100-5+P4 TURN-ON TRANSIENT
20 GPM 150 F

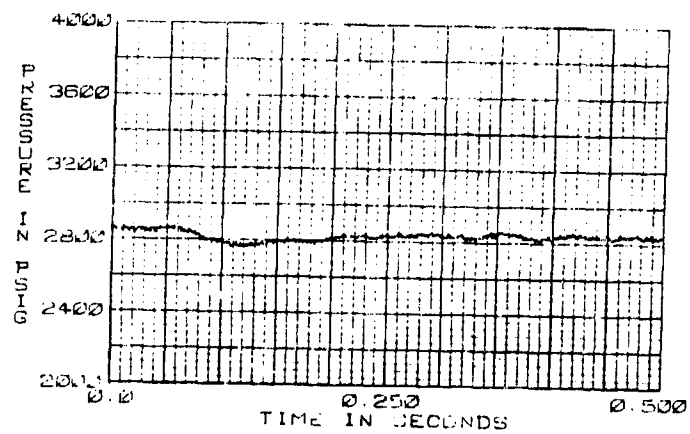


HYTRAN--F-15
100-5+XH TURN-ON TRANSIENT
20 GPM 150 F

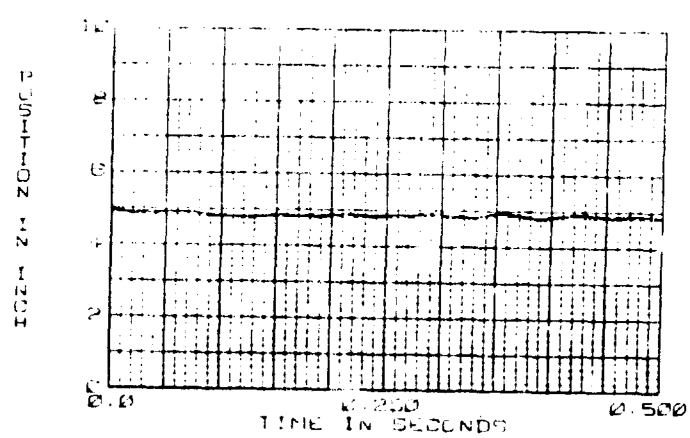
FIGURE 49 F-15 HYDRAULIC PUMP 100-5 TURN-ON TRANSIENT, 77 CIS, 150°F



HYTRAN--F-15
100-5+PS TURN-ON TRANSIENT
20 GPM 150 F



HYTRAN--F-15
100-5+PS TURN-ON TRANSIENT
20 GPM 150 F



HYTRAN--F-15
100-5+XP TURN-ON TRANSIENT
20 GPM 150 F

FIGURE 49 (CONTINUED)

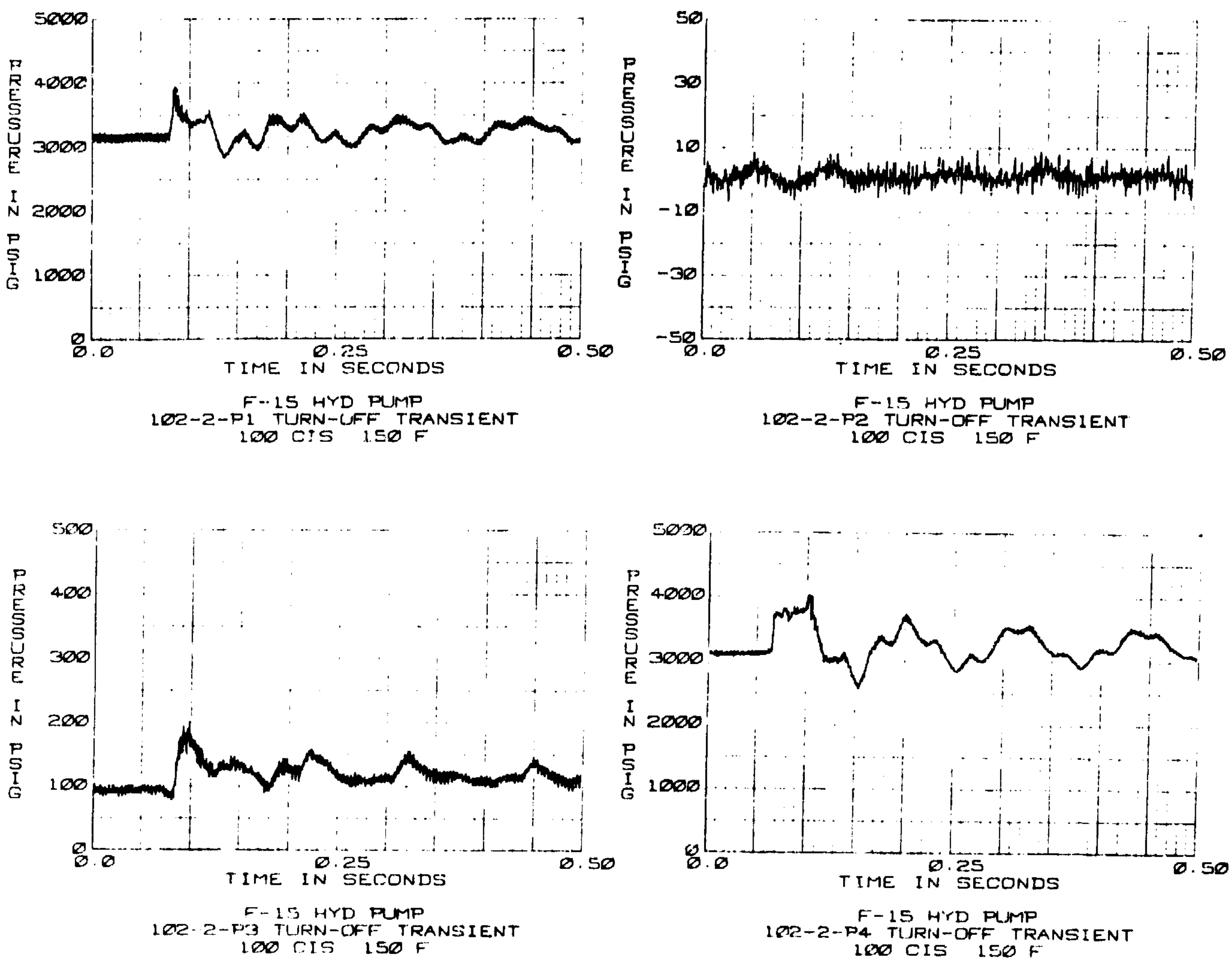
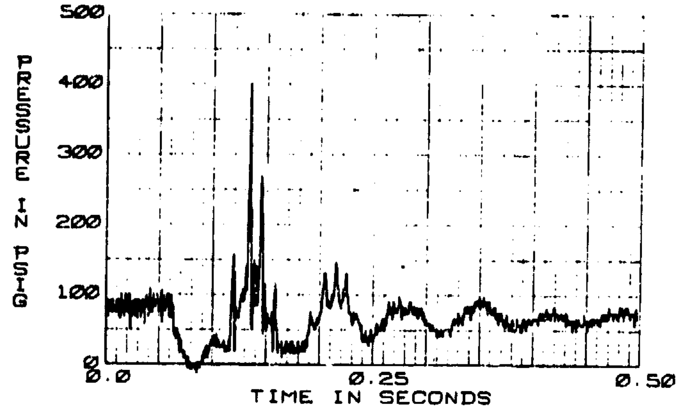
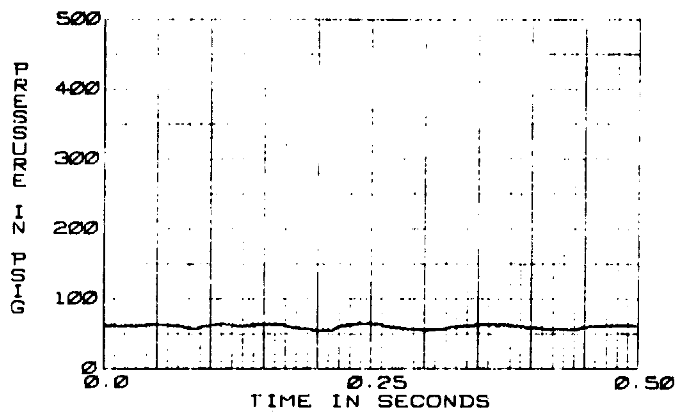


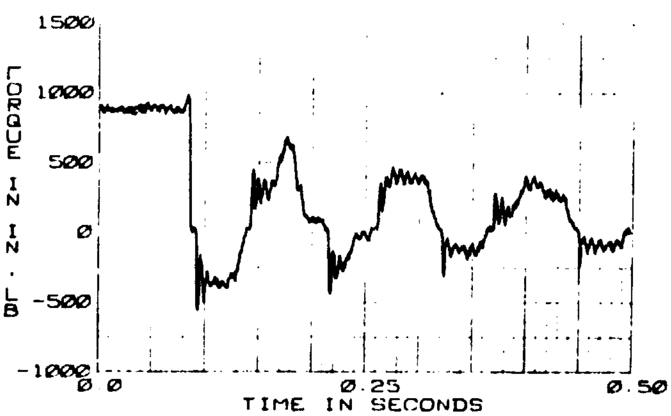
FIGURE 50 F-15 HYDRAULIC PUMP 102-2 TURN-OFF TRANSIENT, 100 CIS, 150°F



F-15 HYD PUMP
102-2-PS TURN-OFF TRANSIENT
100 CIS 150 F



F-15 HYD PUMP
102-2-P7 TURN-OFF TRANSIENT
100 CIS 150 F



F-15 HYD PUMP
102-2-DT TURN-OFF TRANSIENT
100 CIS 150 F

FIGURE 50 (CONTINUED)

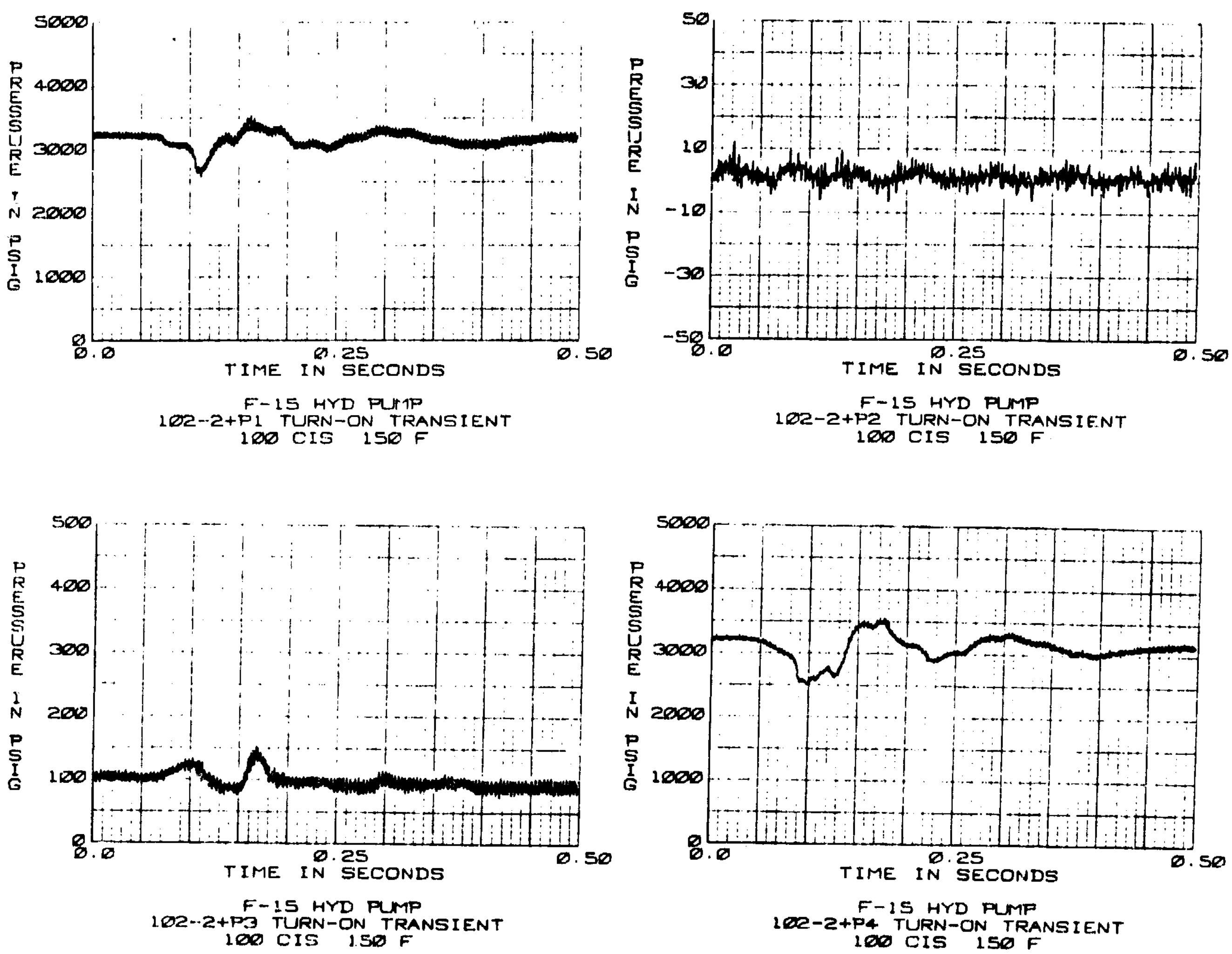
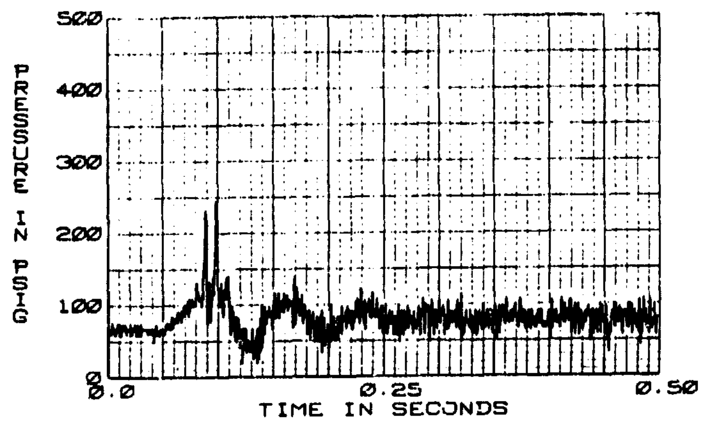
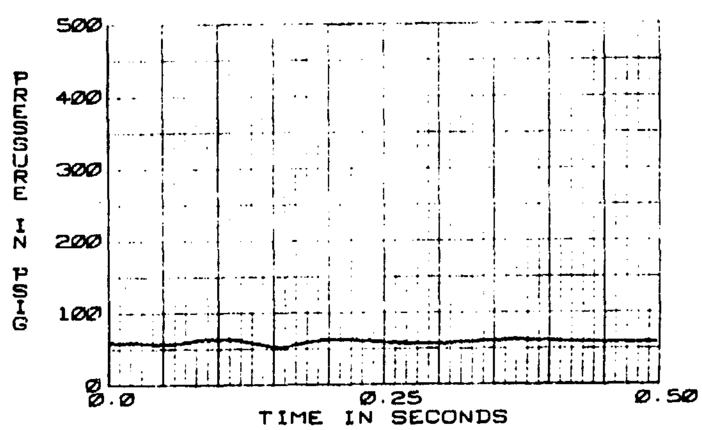


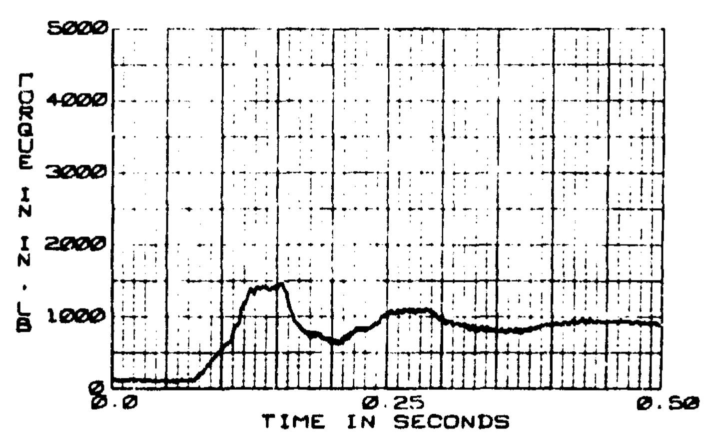
FIGURE 51 F-15 HYDRAULIC PUMP 102-2 TURN-ON TRANSIENT, 100 CIS, 150°F



F-15 HYD PUMP
102-2+P5 TURN-ON TRANSIENT
100 CIS 150 F



F-15 HYD PUMP
102-2+P7 TURN-ON TRANSIENT
100 CIS 150 F



F-15 HYD PUMP
102-2+DT TURN-ON TRANSIENT
100 CIS 150 F

FIGURE 51 (CONTINUED)

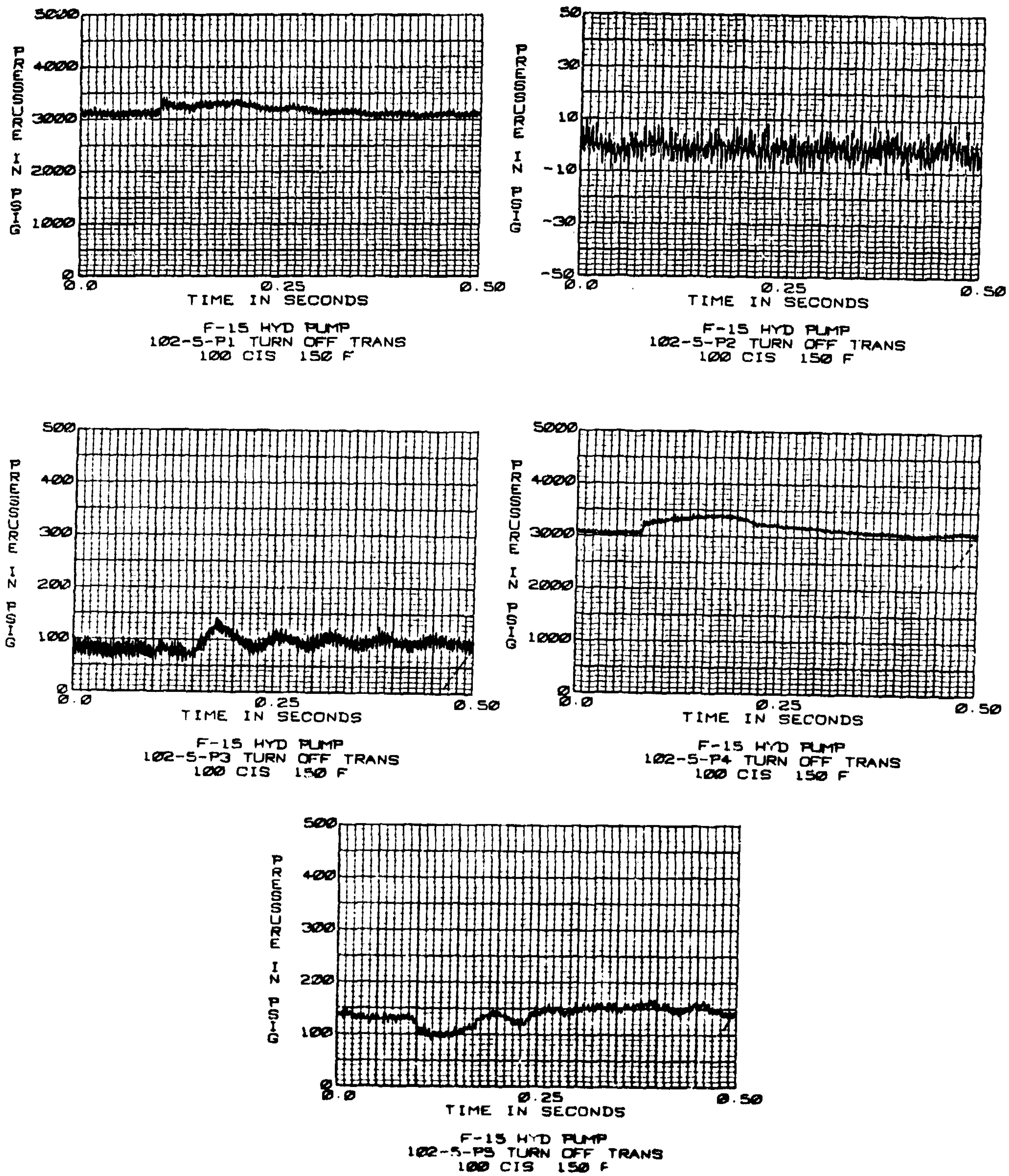


FIGURE 52 F-15 HYDRAULIC PUMP 102-5 TURN-OFF TRANSIENT, 100 CIS, 150°F

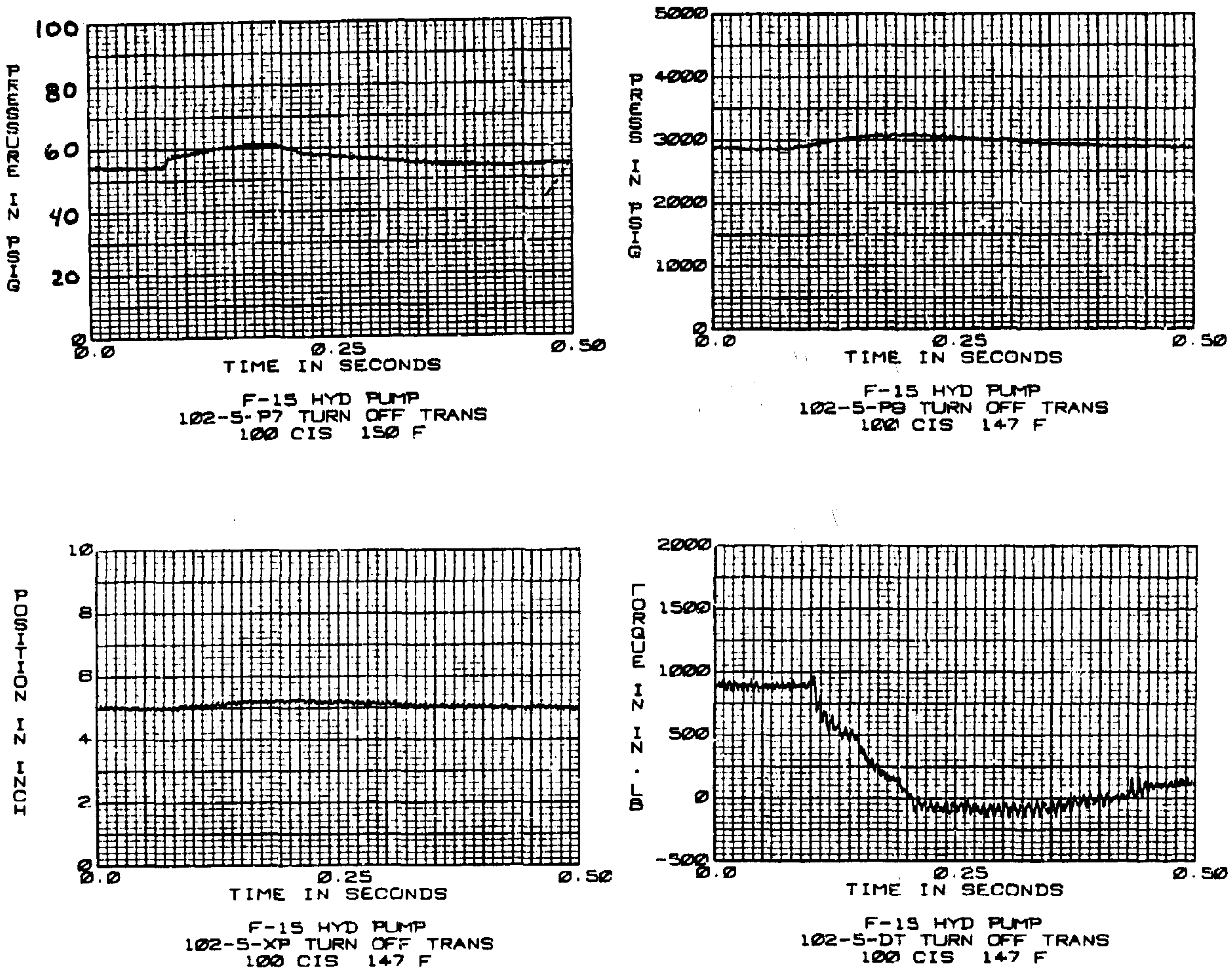


FIGURE 52 (CONTINUED)

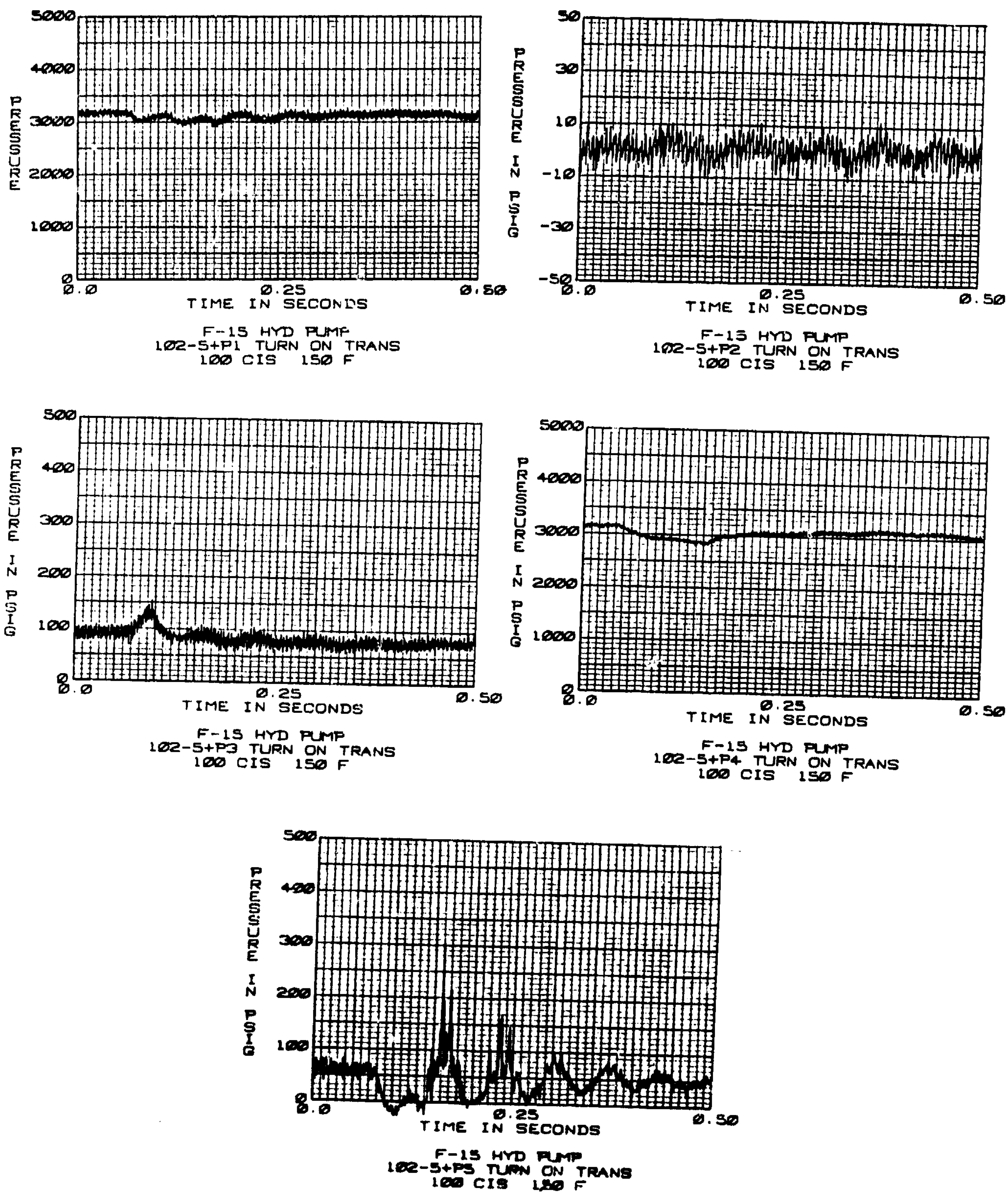
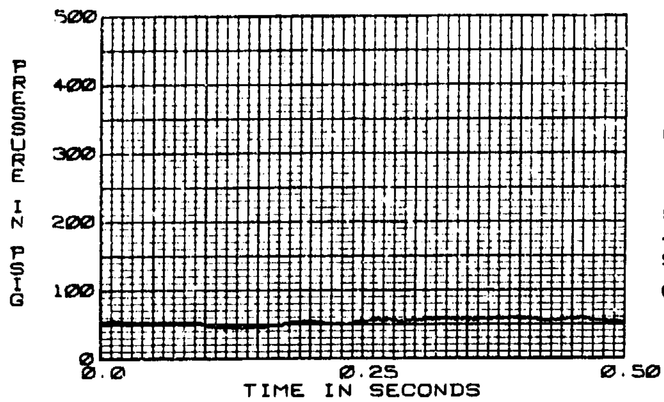
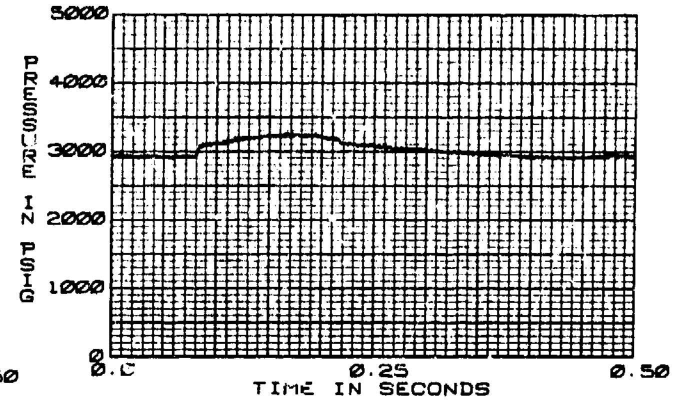


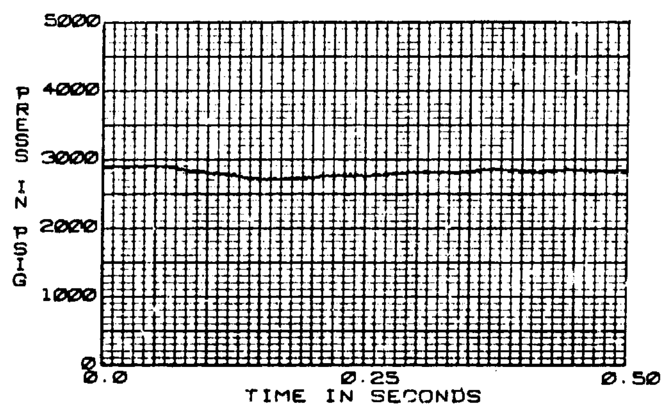
FIGURE 53 F-15 HYDRAULIC PUMP 102-5 TURN-ON TRANSIENT, 100 CIS, 150°F



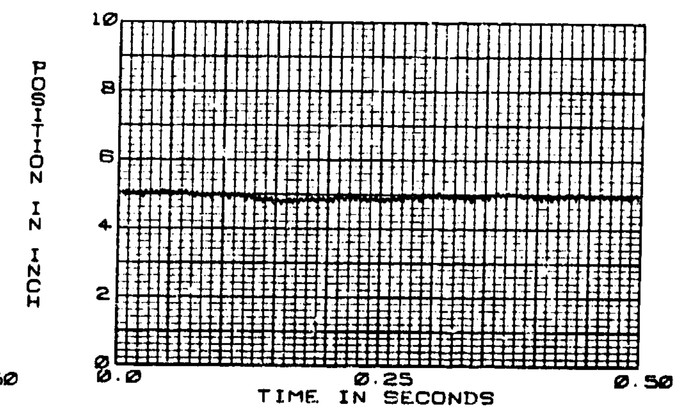
F-15 HYD PUMP
102-5+P7 TURN ON TRANS
100 CIS 150 F



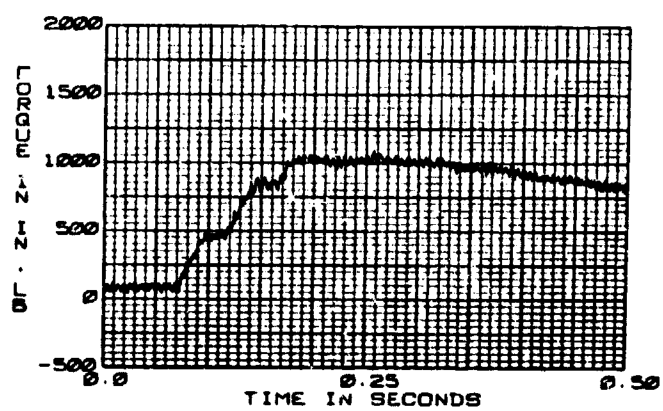
F-15 HYD PUMP
102-5+P8 TURN ON TRANS
100 CIS 150 F



F-15 HYD PUMP
102-5+P9 TURN ON TRANS
100 CIS 150 F



F-15 HYD PUMP
102-5+XP TURN ON TRANS
100 CIS 150 F



F-15 HYD PUMP
102-5+DT TURN ON TRANS
100 CIS 150 F

FIGURE 53 (CONTINUED)

(d) F-4 Pump Transient Testing 31.3 in^3 System

Transient test runs were made at 10, 50, 77 and 104 CIS pump steady state flows with and without the JFS accumulator in the test system. Results for the 77 CIS turn-off and turn-on transients are presented in Figures 54 and 55. When the accumulator was added, the data in Figures 56 and 57 resulted.

(e) F-4 Pump Transient Testing 270.66 in^3 System

Turn-on and turn-off transient runs were made for steady state flows of 10, 50, 77 and 104 CIS. Transient test results for the 77 CIS flowrate are shown on Figures 58 and 59. The same steady state flow runs with the JFS accumulator are presented in Figures 60 and 61.

(f) F-4 Pump Transient Testing 538.1 in^3 System

Turn-off and turn-on transients were run with the F-4 pump at steady state flows of 10, 50, 77 and 104 CIS. Test data for the 104 CIS run are shown in Figures 62 and 63. Test data with the accumulator added to the test circuit are presented in Figures 64 and 65.

The test system load was not large enough to show significant changes in the JFS accumulator gas and oil pressures and piston position. The fluid inertance effects could not be determined. The change in pressure between the line and accumulator were rapid and small in magnitude. The rate of change in flow as measured by differentiating the piston position was also rapid. When the pressure change is applied too quickly, compressibility effects cannot be ignored and the flowrate is not uniform. Consequently, transient accumulator performance was not simulated with the empirical pump models.

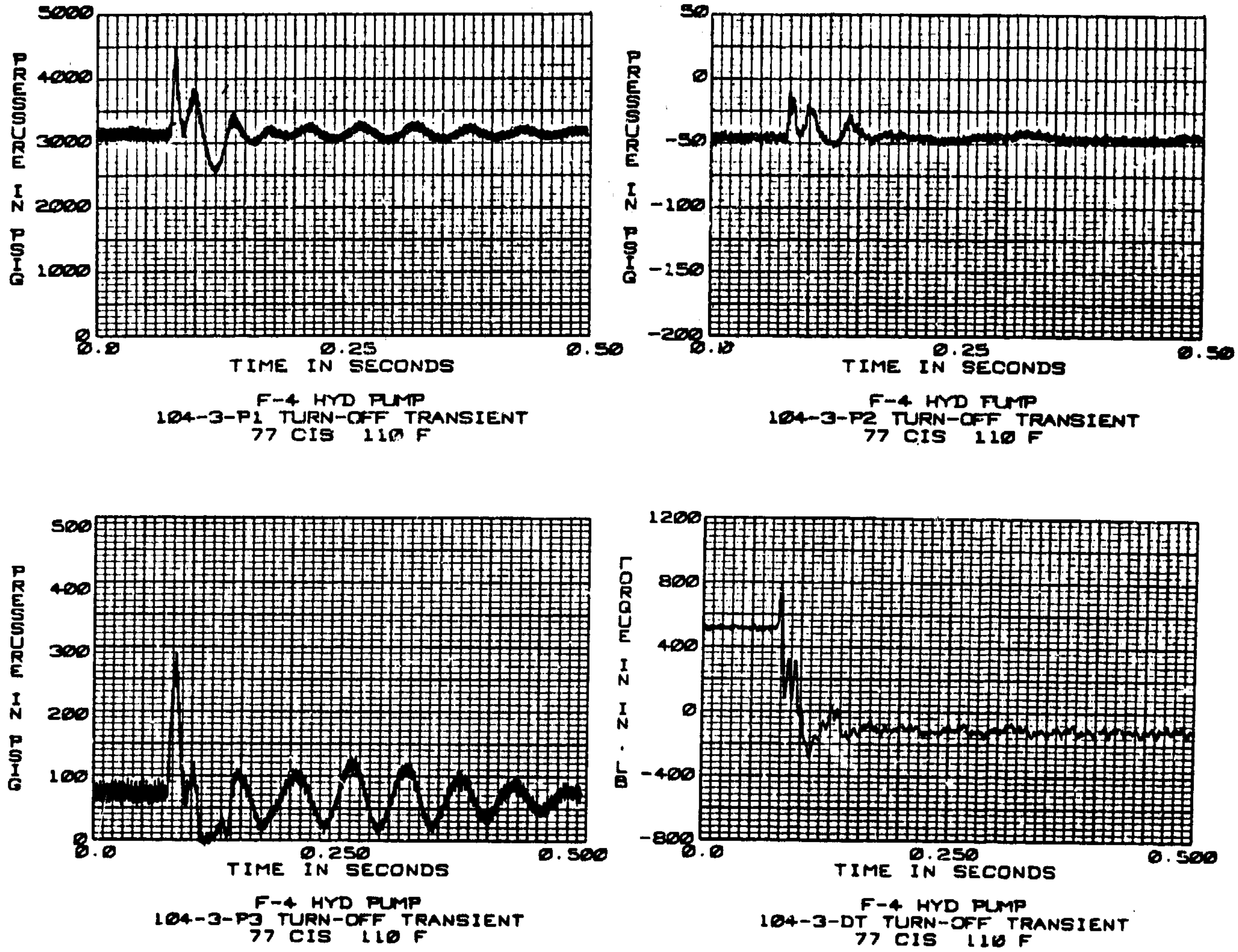


FIGURE 54 F-4 HYDRAULIC PUMP 104-3 TURN-OFF TRANSIENT, 77 CIS, 110°F

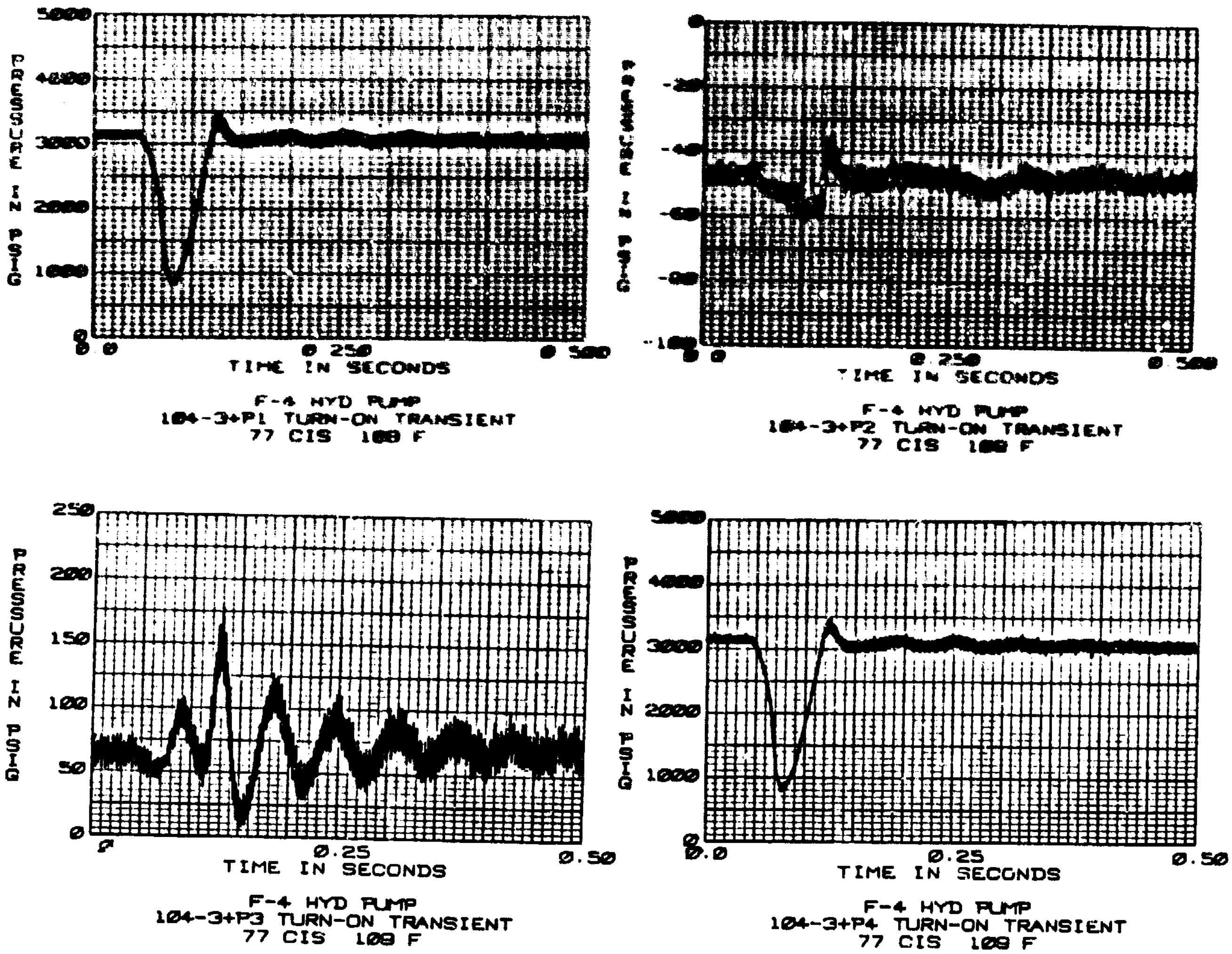
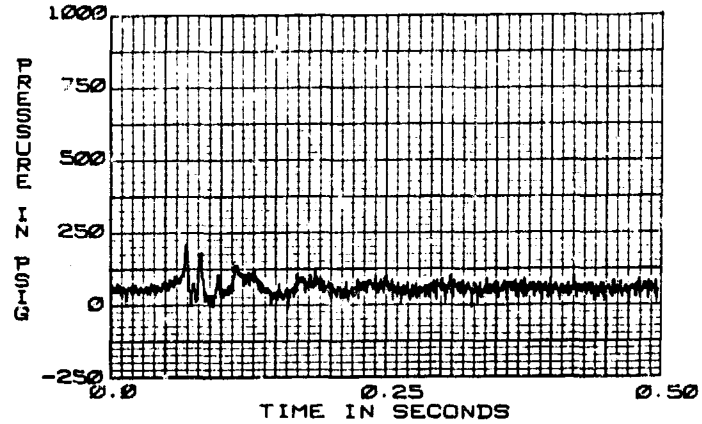
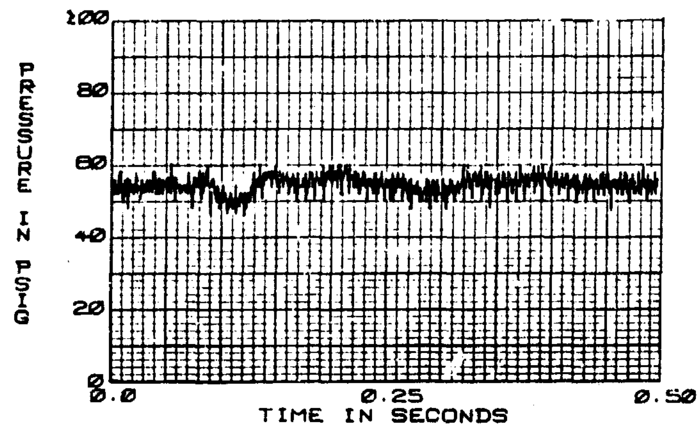


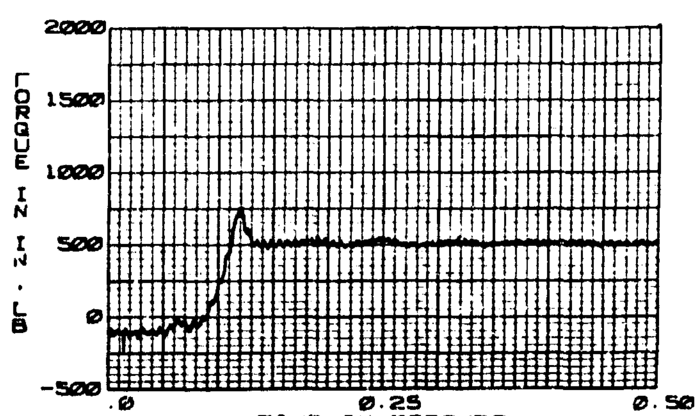
FIGURE 55 F-4 HYDRAULIC PUMP 104-3 TURN-ON TRANSIENT, 77 CIS, 109°F



F-4 HYD PUMP
104-3+P5 TURN-ON TRANSIENT
77 CIS 108 F



F-4 HYD PUMP
104-3+P7 TURN-ON TRANSIENT
77 CIS 108 F



YES
104-3+DT TURN-ON TRANSIENT
77 CIS 108 F

FIGURE 55 (CONTINUED)

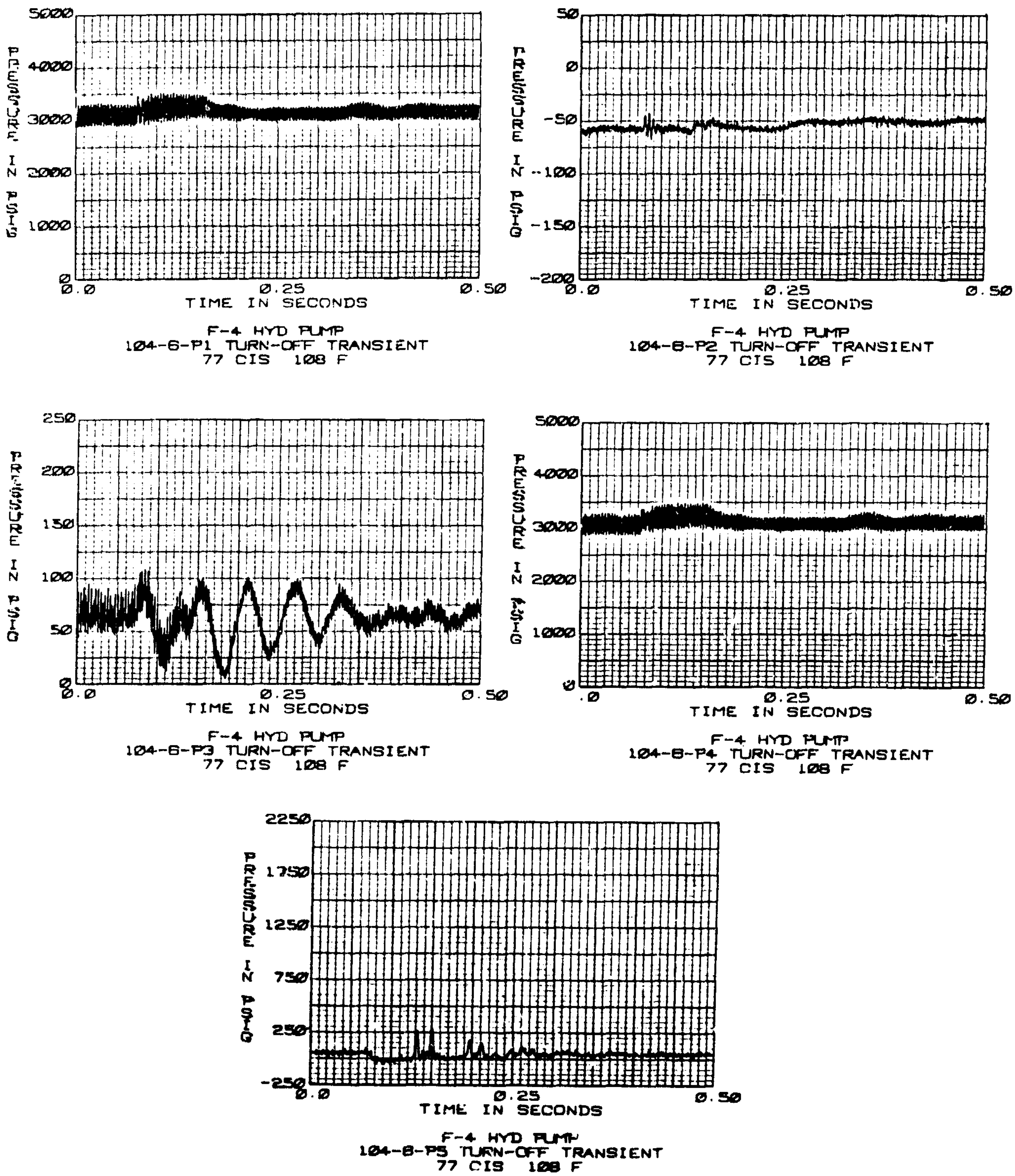
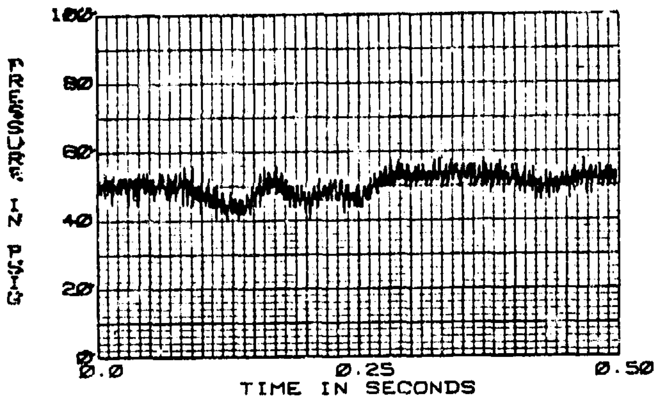
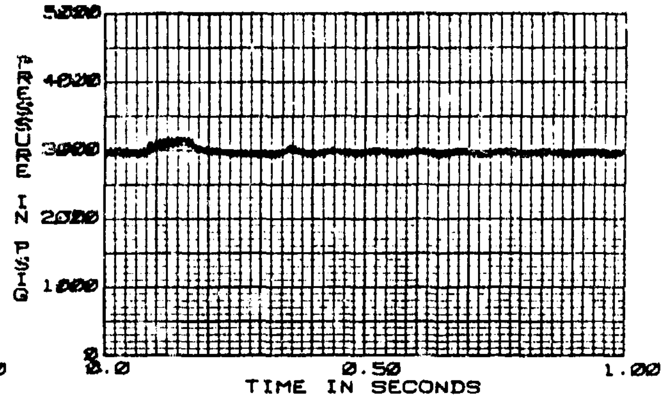


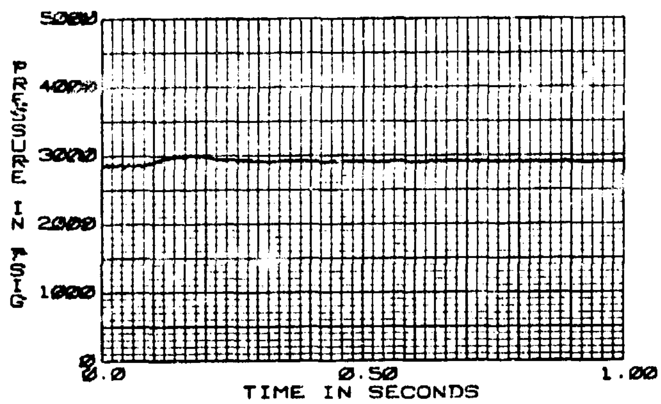
FIGURE 56 F-4 HYDRAULIC PUMP 104-6 TURN-OFF TRANSIENT, 77 CIS, 108°F



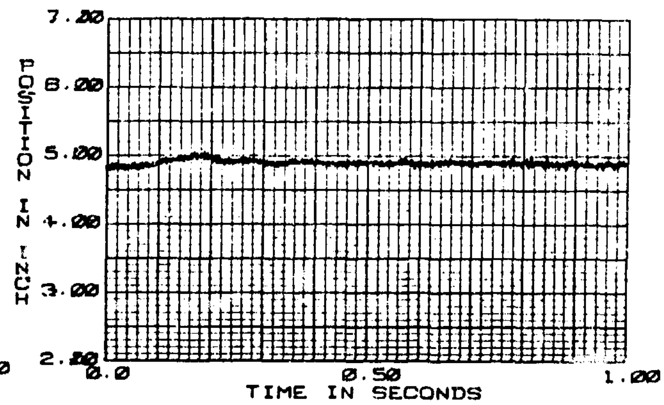
F-4 HYD PUMP
104-6-P7 TURN-OFF TRANSIENT
77 CIS 108 F



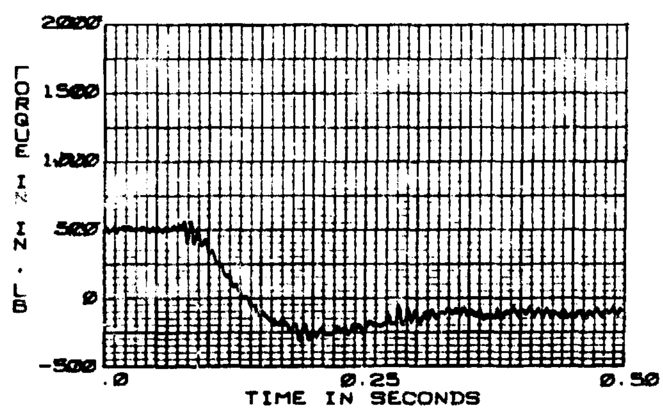
F-4 HYD PUMP
104-6-P8 TURN-OFF TRANSIENT
77 CIS 108 F



F-4 HYD PUMP
104-6-P9 TURN-OFF TRANSIENT
77 CIS 108 F



F-4 HYD PUMP
104-6-XP TURN-OFF TRANSIENT
77 CIS 108 F



F-4 HYD PUMP
104-6-DT TURN-OFF TRANSIENT
77 CIS 108 F

FIGURE 56 (CONTINUED)

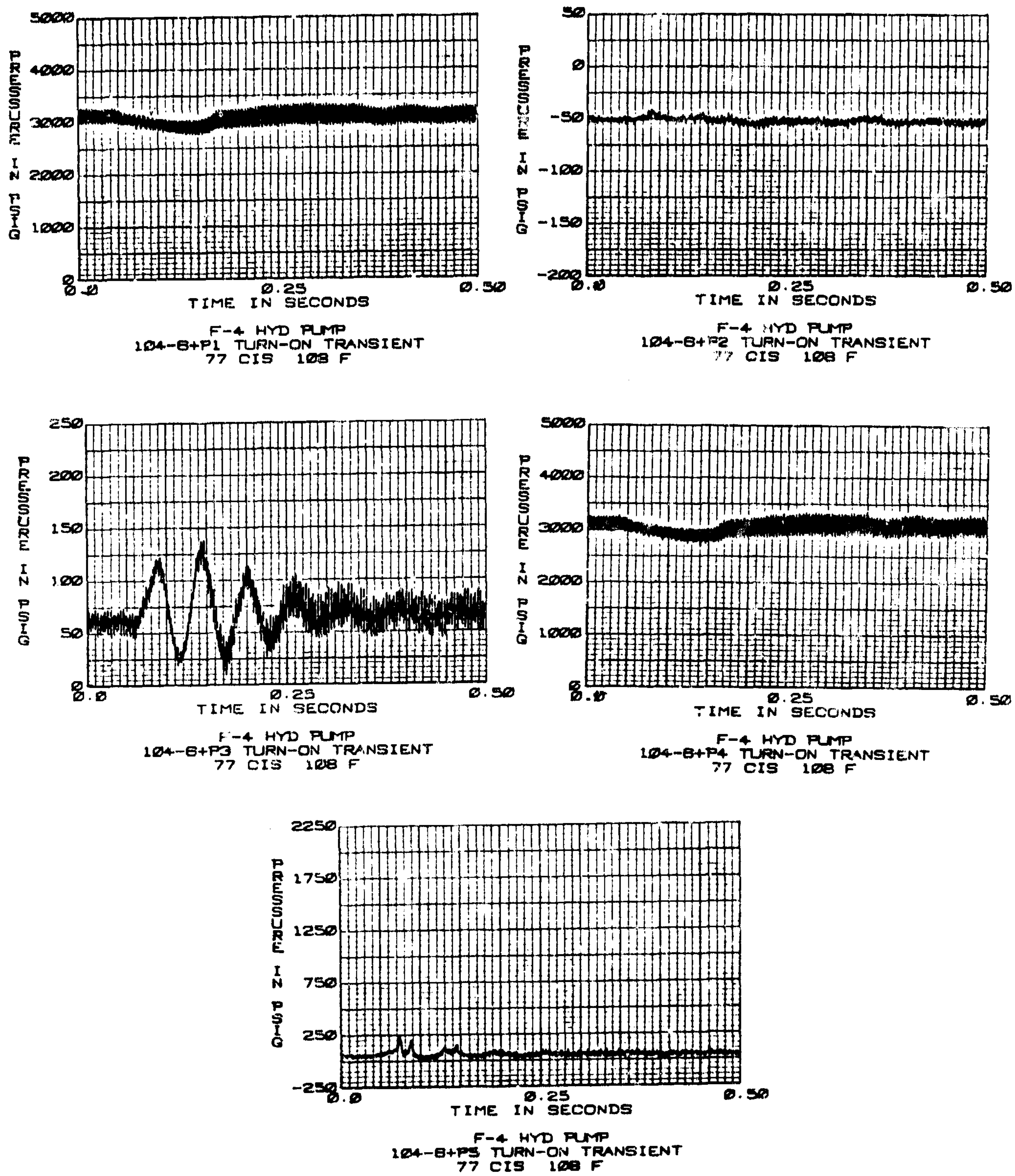
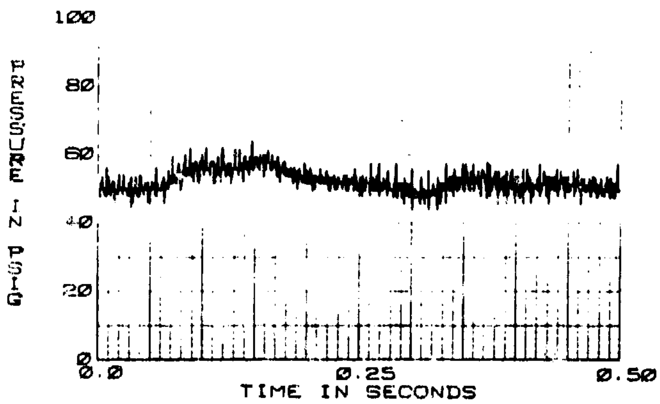
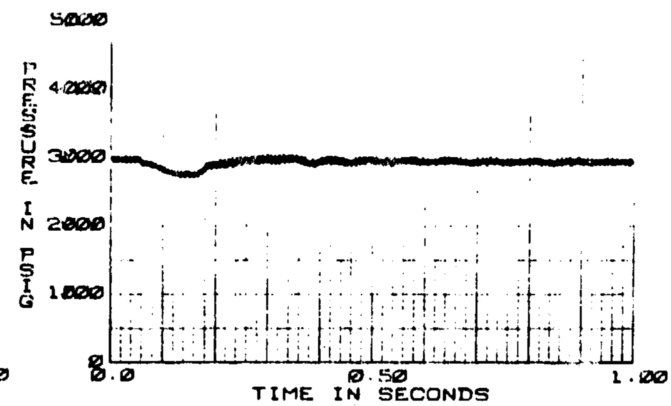


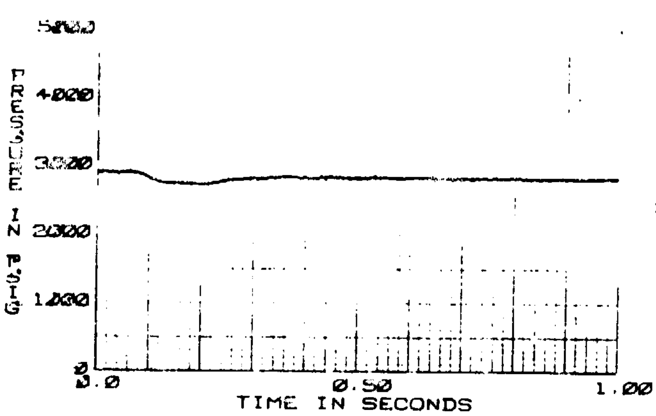
FIGURE 57 F-4 HYDRAULIC PUMP 104-6 TURN-ON TRANSIENT, 77 CIS, 108°F



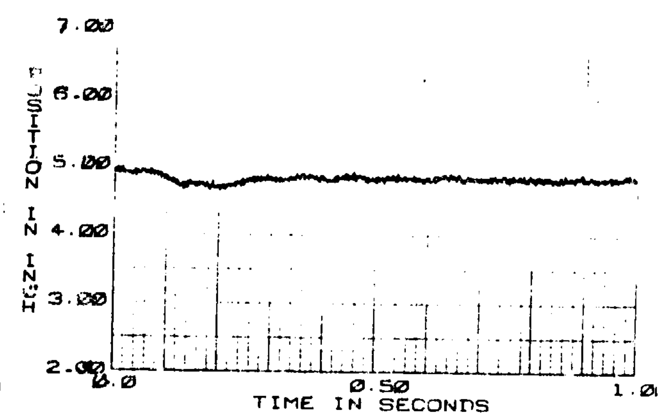
F-4 HYD PUMP
104-6+P7 TURN-ON TRANSIENT
77 CIS 108 F



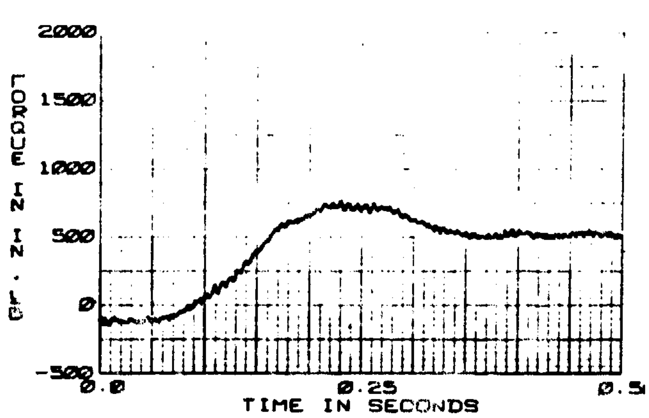
F-4 HYD PUMP
104-6+P8 TURN-ON TRANSIENT
77 CIS 108 F



F-4 HYD PUMP
104-6+P9 TURN-ON TRANSIENT
77 CIS 108 F



F-4 HYD PUMP
104-6+XP TURN-ON TRANSIENT
77 CIS 108 F



F-4 HYD PUMP
104-6+DT TURN-ON TRANSIENT
77 CIS 108 F

FIGURE 57 (CONTINUED)

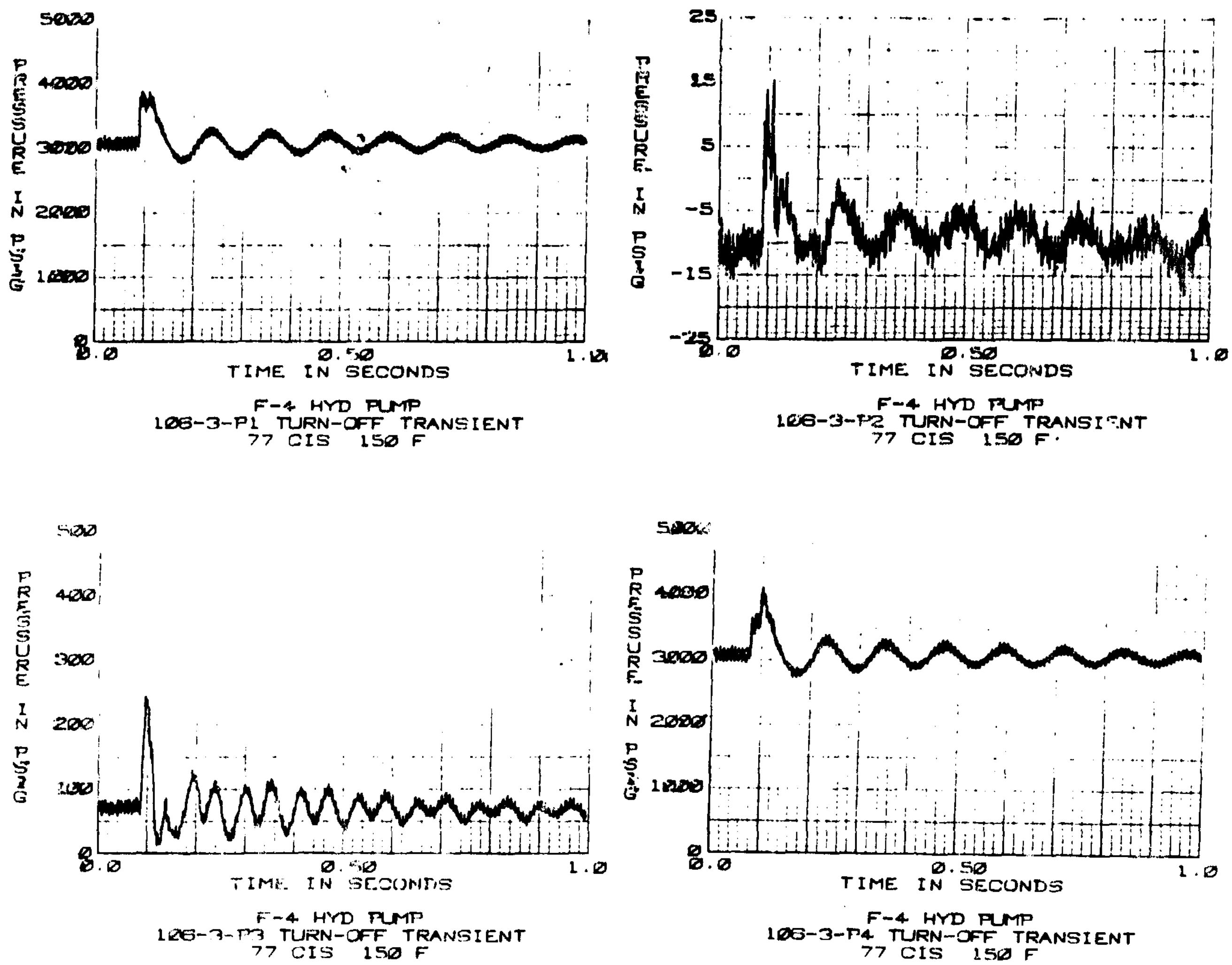
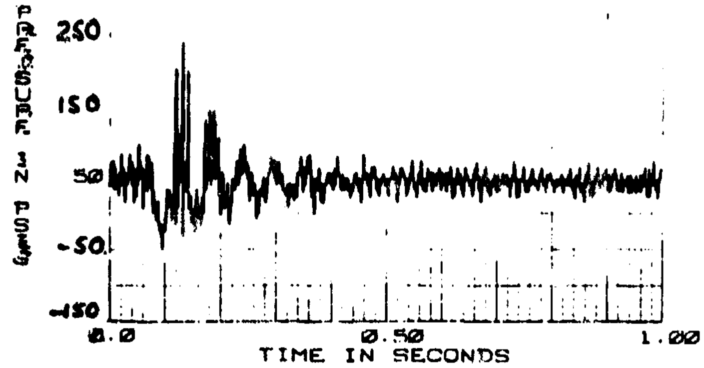
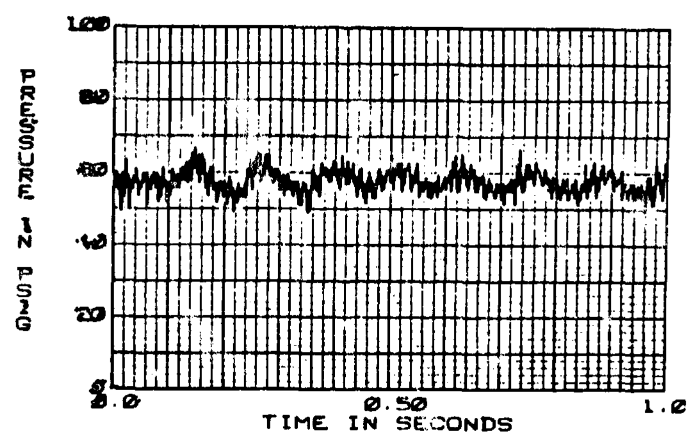


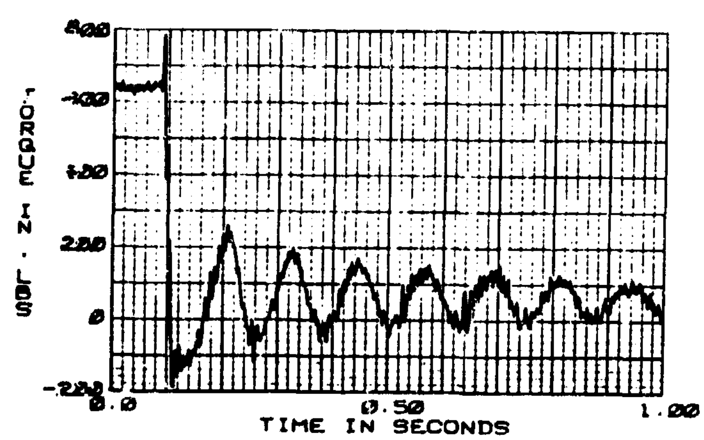
FIGURE 58 F-4 HYDRAULIC PUMP 106-3 TURN-OFF TRANSIENT, 77 CIS, 150°F



F-4 HYD PUMP
108-3-P5 TURN-OFF TRANSIENT
77 CIS 150 F



F-4 HYD PUMP
106-3-P7 TURN-OFF TRANSIENT
77 CIS 150 F



F-4 HYD PUMP
108-3-DT TURN-OFF TRANSIENT
77 CIS 150 F

FIGURE 58 (CONTINUED)

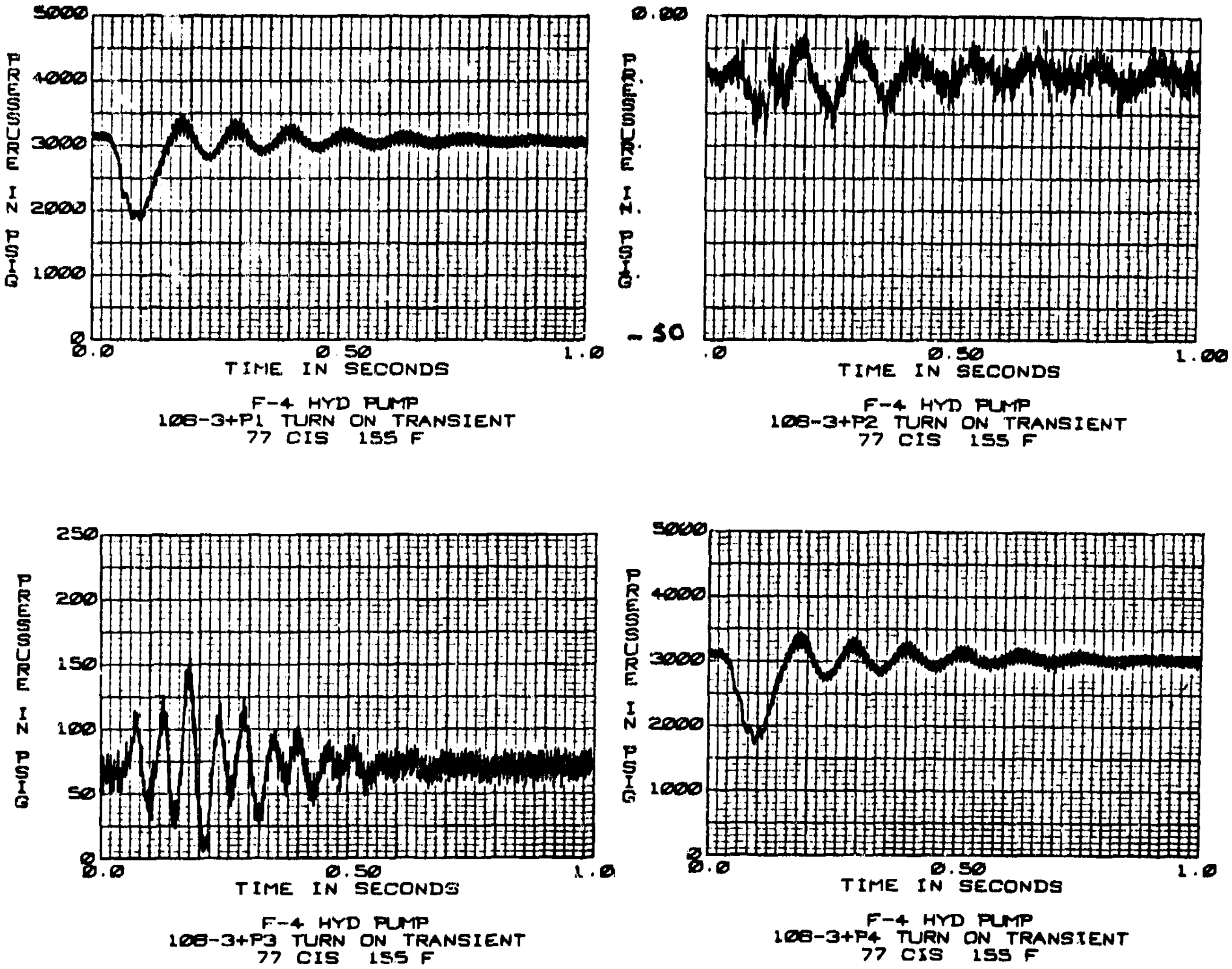
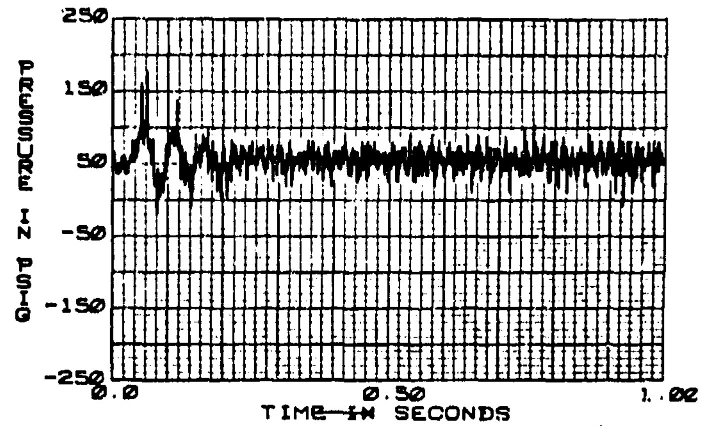
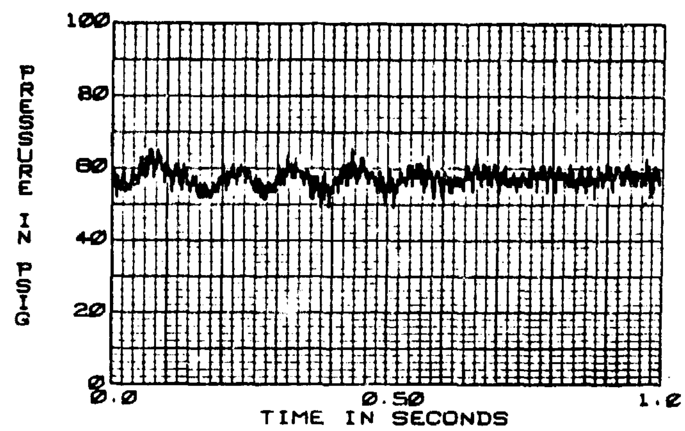


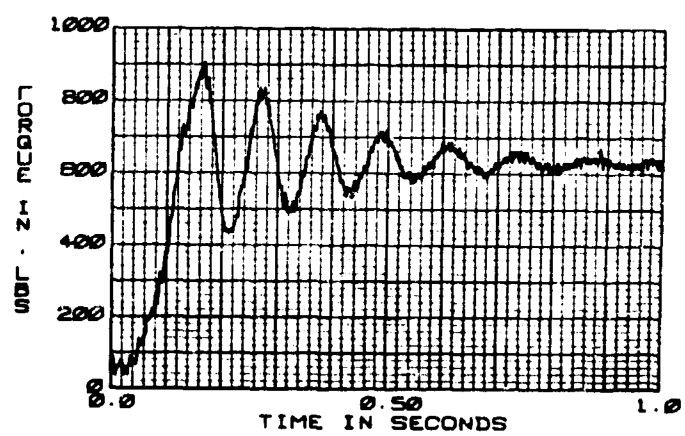
FIGURE 59 F-4 HYDRAULIC PUMP 106-3 TURN-ON TRANSIENT, 77 CIS, 155°F



F-4 HYD PUMP
106-3+P5 TURN ON TRANSIENT
77 CIS 155 F



F-4 HYD PUMP
106-3+P7 TURN ON TRANSIENT
77 CIS 155 F



F-4 HYD PUMP
106-3+DT TURN ON TRANSIENT
77 CIS 155 F

FIGURE 59 (CONTINUED)

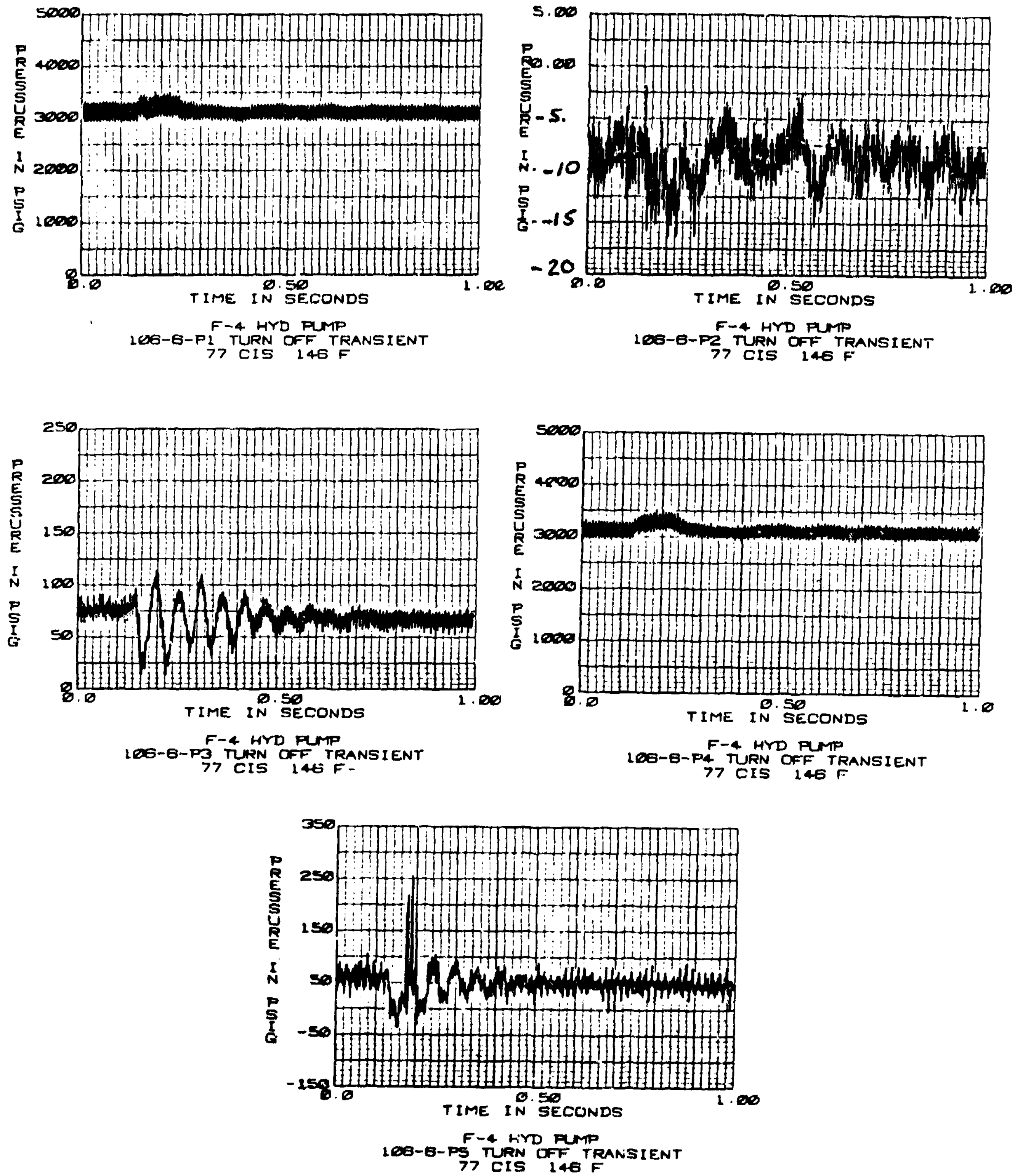
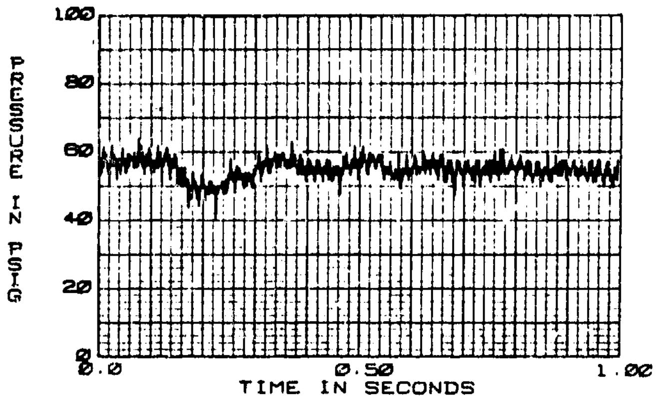
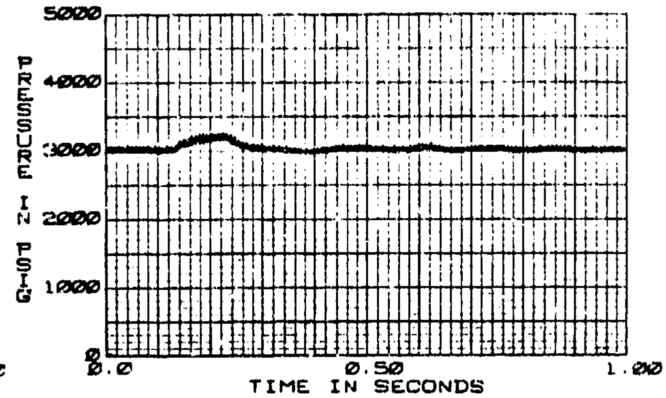


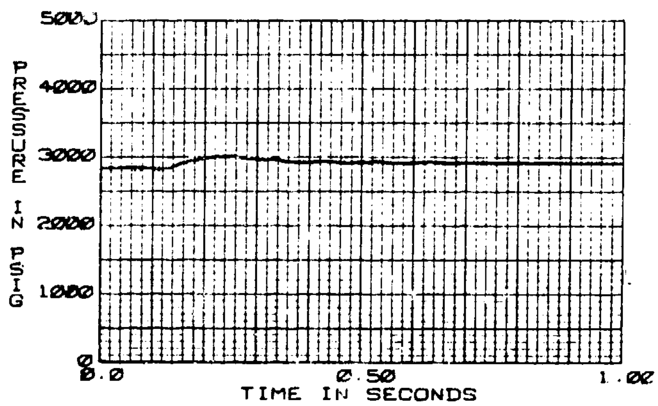
FIGURE 60 F-4 HYDRAULIC PUMP 106-6 TURN-OFF TRANSIENT, 77 CIS, 146°F



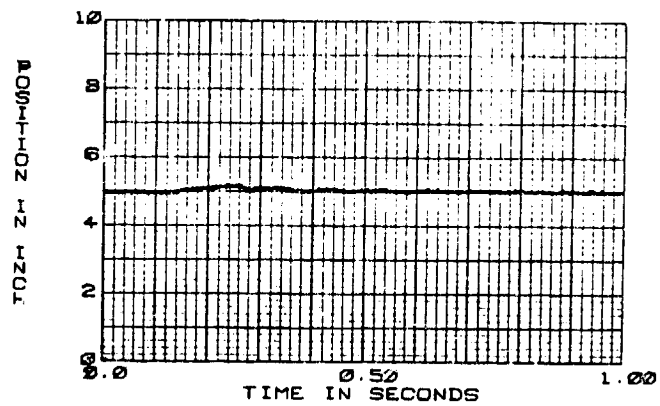
F-4 HYD PUMP
106-6-P7 TURN OFF TRANSIENT
77 CIS 146 F



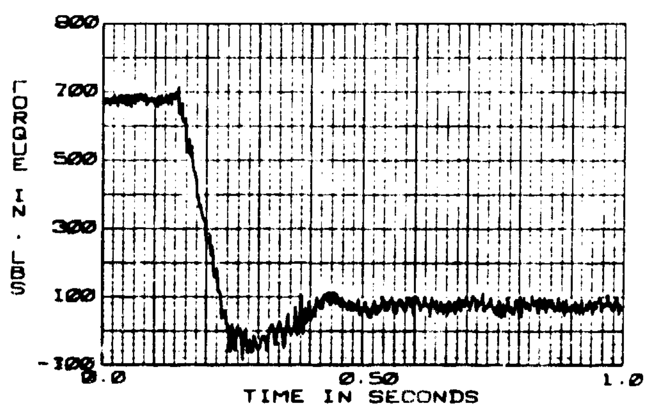
F-4 HYD PUMP
106-6-P8 TURN OFF TRANSIENT
77 CIS 146 F



F-4 HYD PUMP
106-6-P9 TURN OFF TRANSIENT
77 CIS 146 F



F-4 HYD PUMP
106-6-XP TURN OFF TRANSIENT
77 CIS 146 F



F-4 HYD PUMP
106-6-DT TURN OFF TRANSIENT
77 CIS 146 F

FIGURE 60 (CONTINUED)

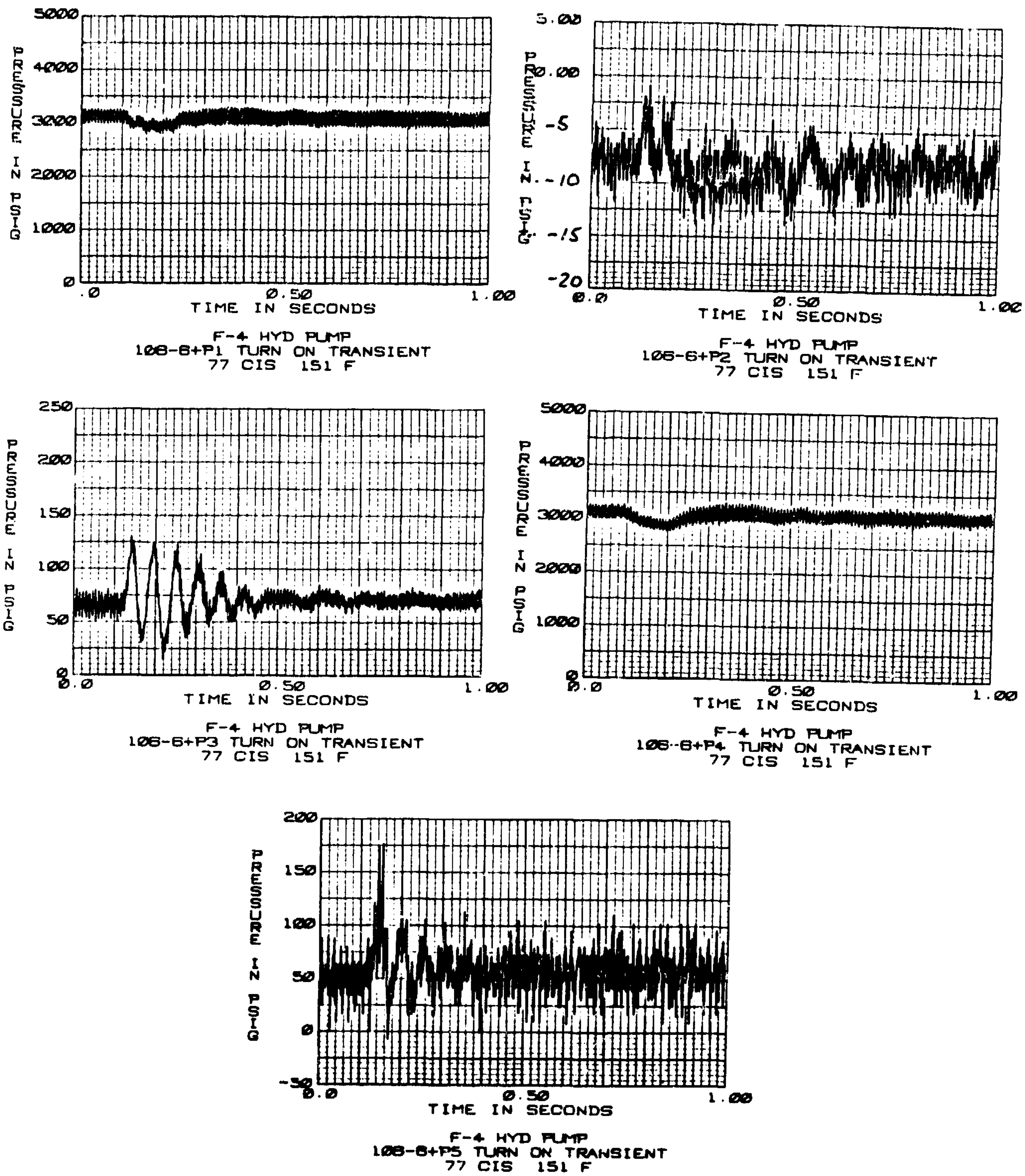
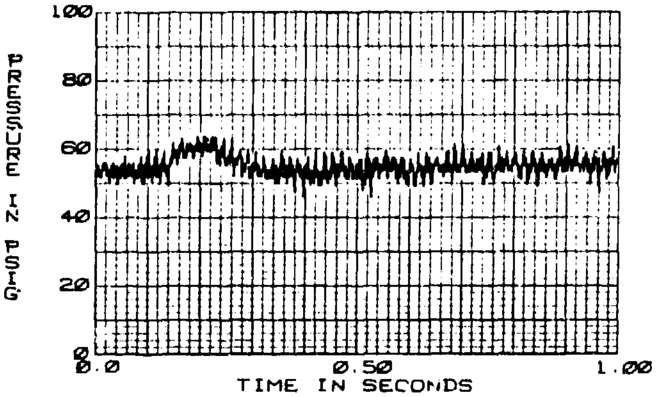
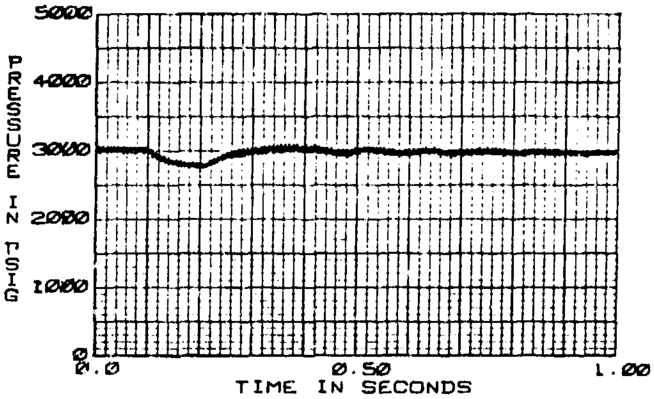


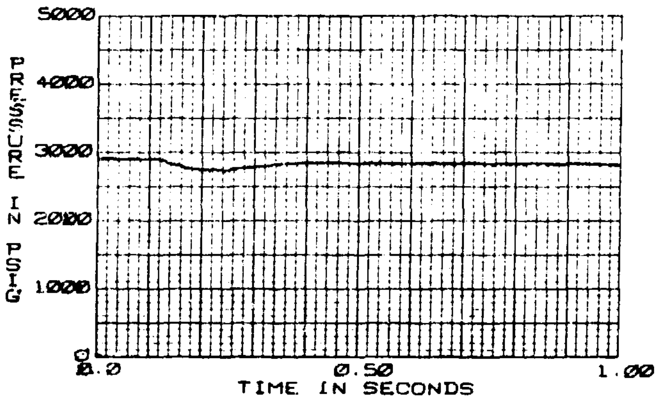
FIGURE 61 F-4 HYDRAULIC PUMP 106-6 TURN-ON TRANSIENT, 77 CIS, 151°F



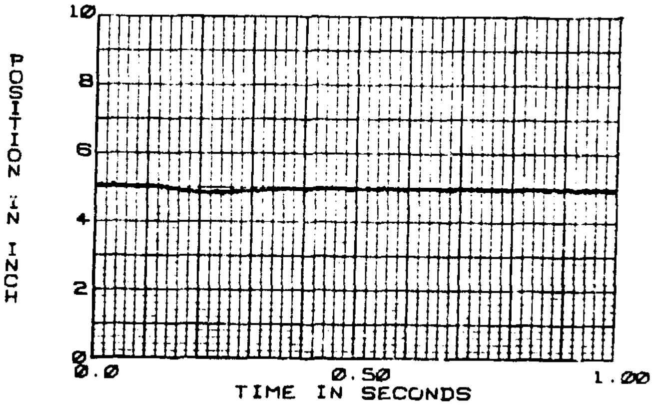
F-4 HYD PUMP
106-6+P7 TURN ON TRANSIENT
77 CIS 151 F



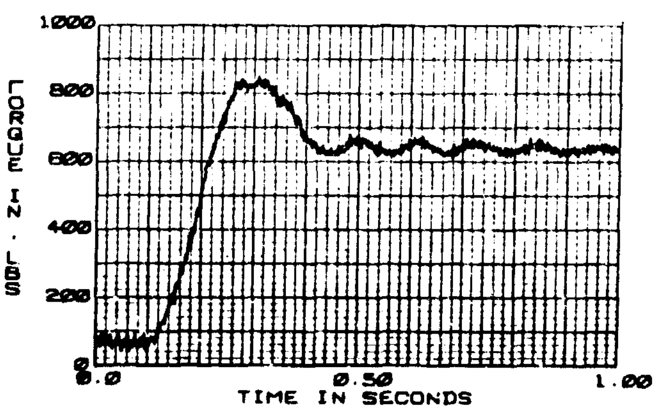
F-4 HYD PUMP
106-6+P8 TURN ON TRANSIENT
77 CIS 151 F



F-4 HYD PUMP
106-6+P9 TURN ON TRANSIENT
77 CIS 151 F



F-4 HYD PUMP
106-6+XP TURN ON TRANSIENT
77 CIS 151 F



F-4 HYD PUMP
106-6+DT TURN ON TRANSIENT
77 CIS 151 F

FIGURE 61 (CONTINUED)

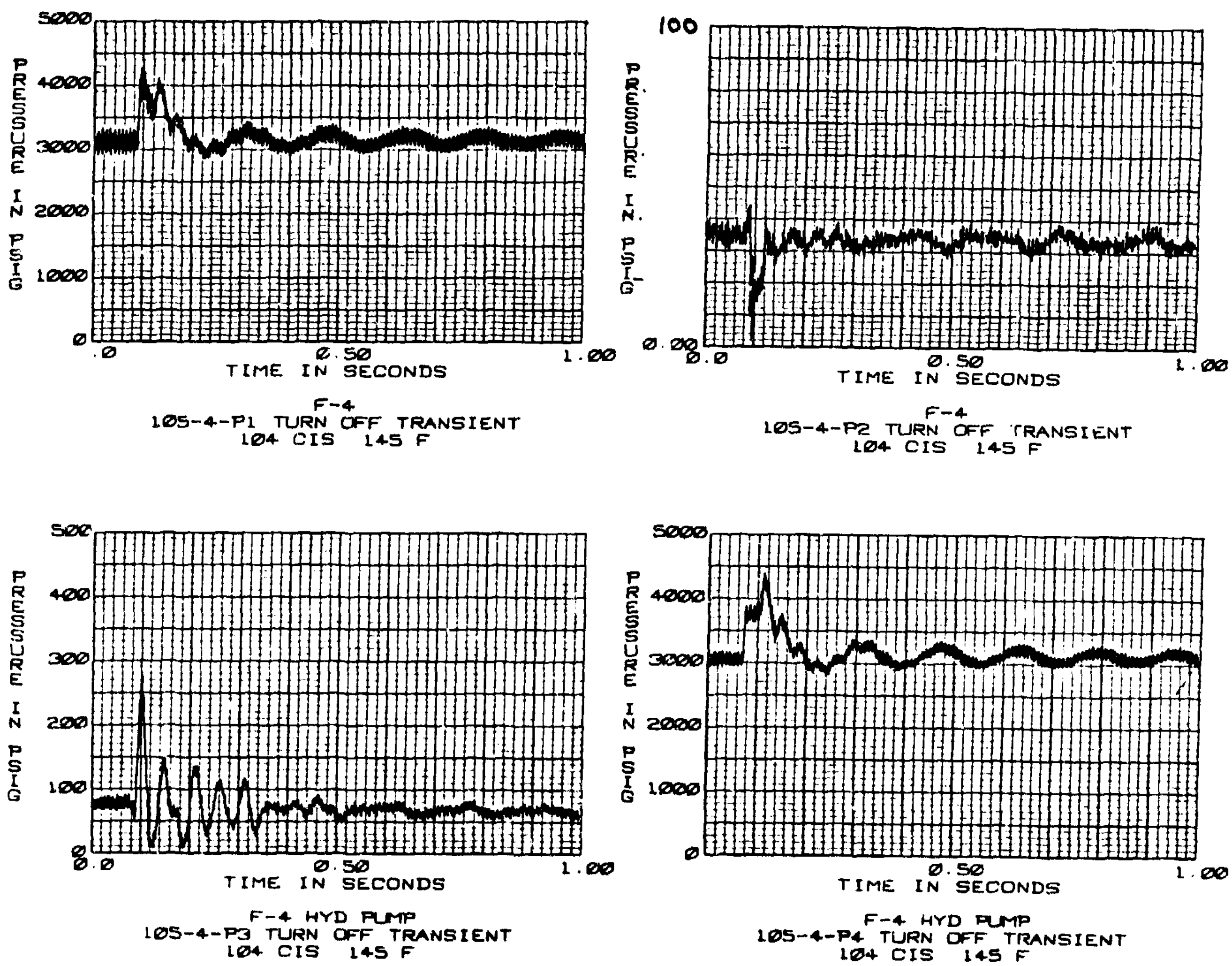
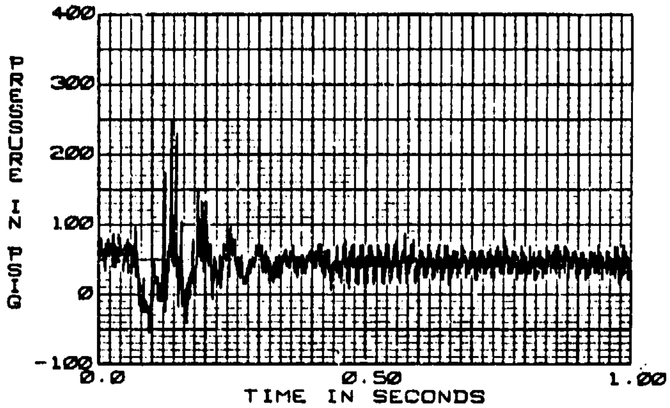
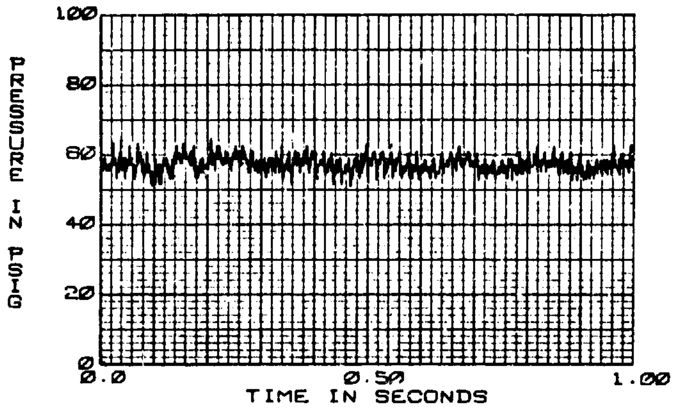


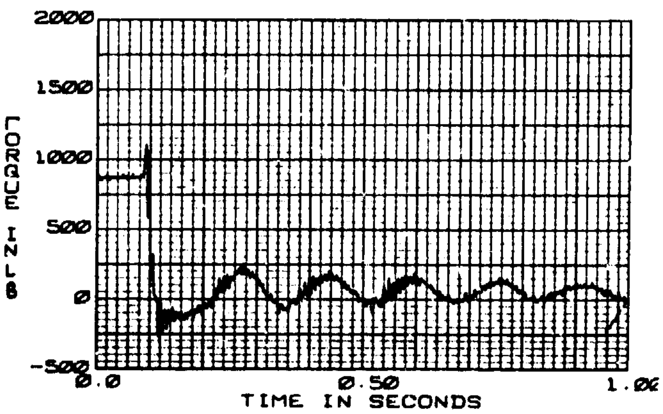
FIGURE 62 F-4 HYDRAULIC PUMP 105-4 TURN-OFF TRANSIENT, 104 CIS, 145°F



F-4 HYD PUMP
105-4-P5 TURN OFF TRANSIENT
104 CIS 145 F



F-4 HYD PUMP
105-4-P7 TURN OFF TRANSIENT
104 CIS 145 F



F-4 HYD PUMP
105-4-DT TURN OFF TRANSIENT
104 CIS 145 F

FIGURE 62 (CONTINUED)

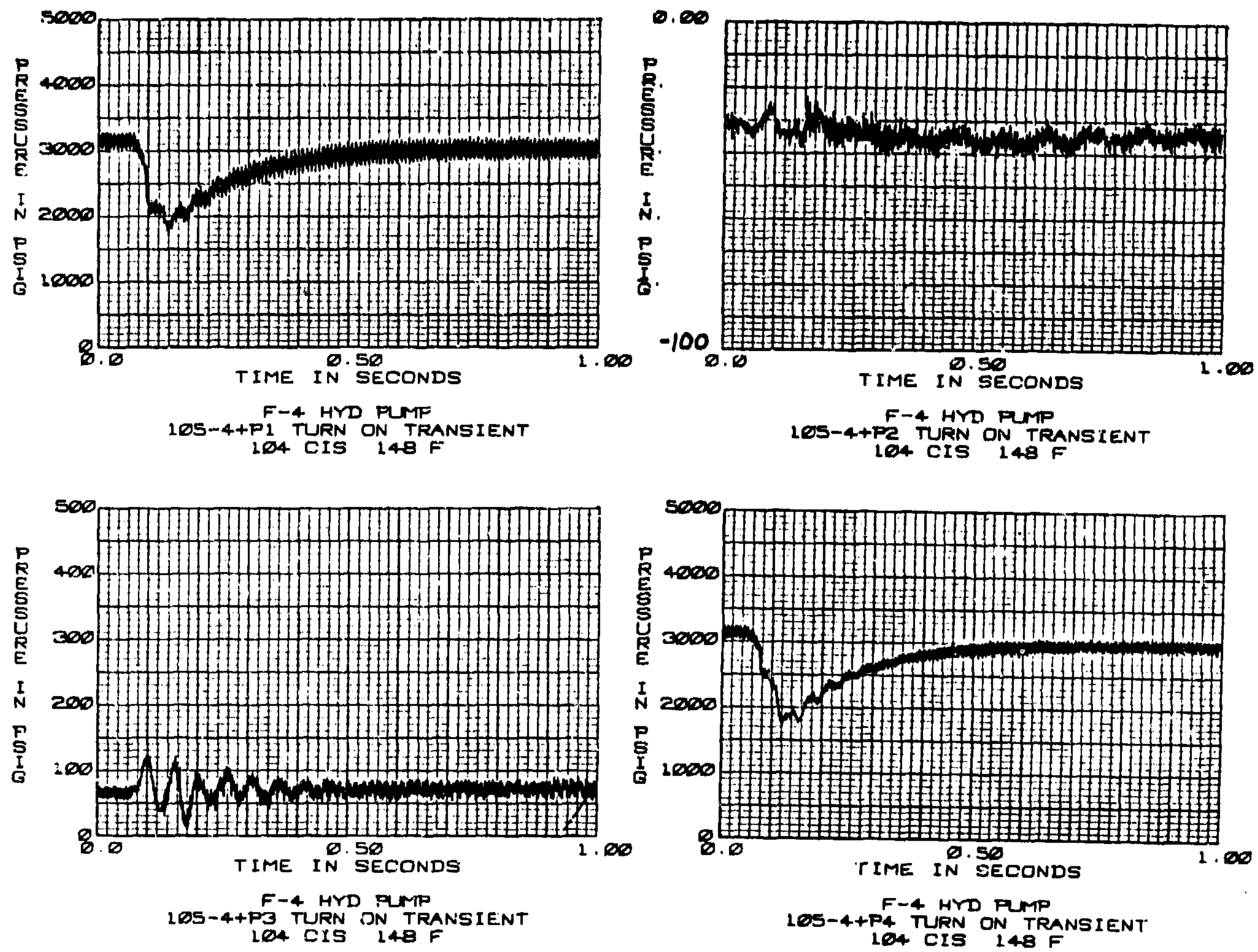
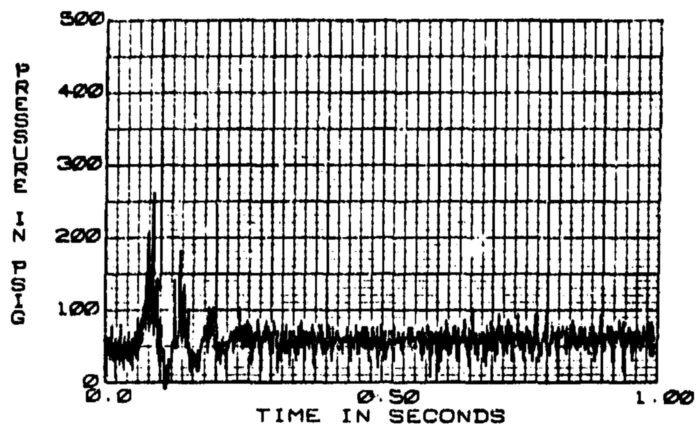
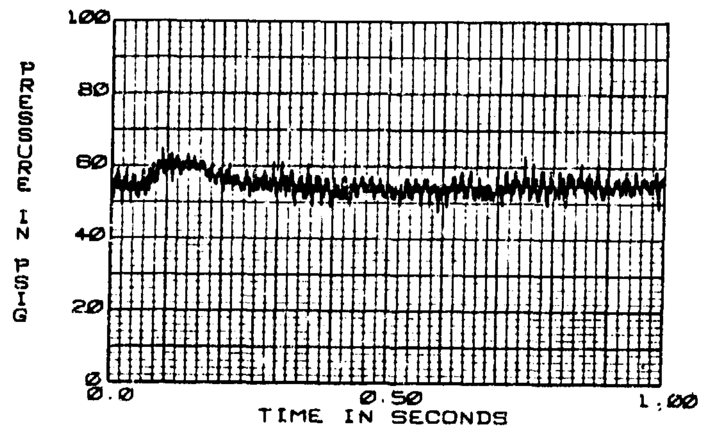


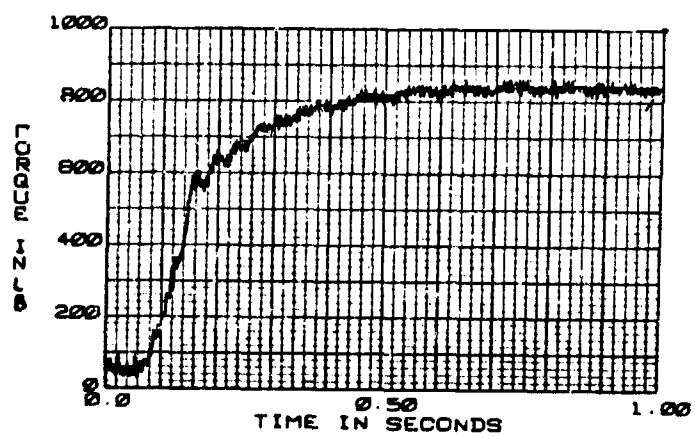
FIGURE 63 F-4 HYDRAULIC PUMP 105-4 TURN-ON TRANSIENT, 104 CIS, 148°F



F-4 HYD PUMP
105-4+P5 TURN ON TRANSIENT
104 CIS 148 F



F-4 HYD PUMP
105-4+P7 TURN ON TRANSIENT
104 CIS 148 F



F-4 HYD PUMP
105-4+DT TURN ON TRANSIENT
104 CIS 148 F

FIGURE 63 (CONTINUED)

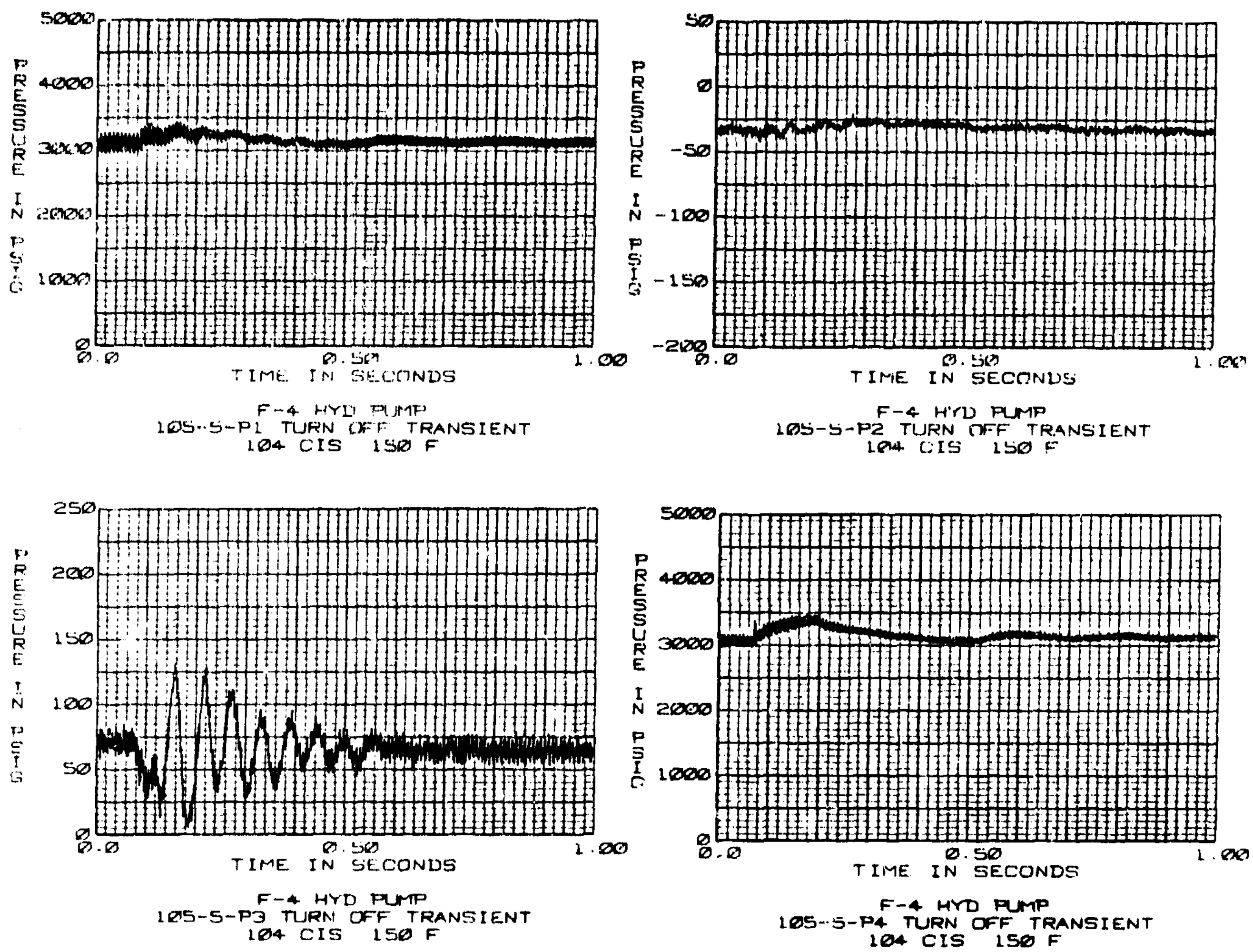
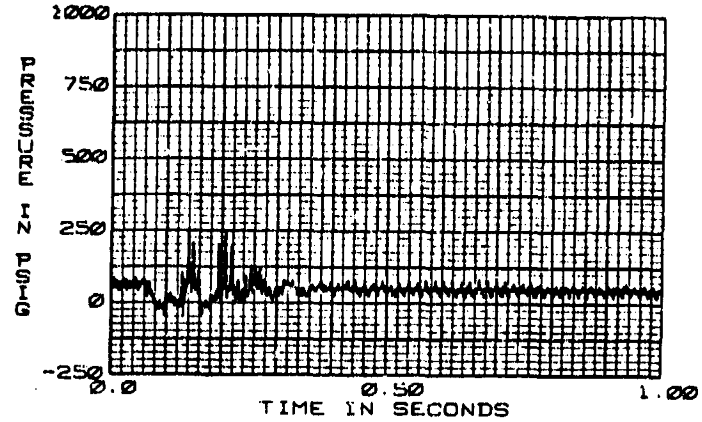
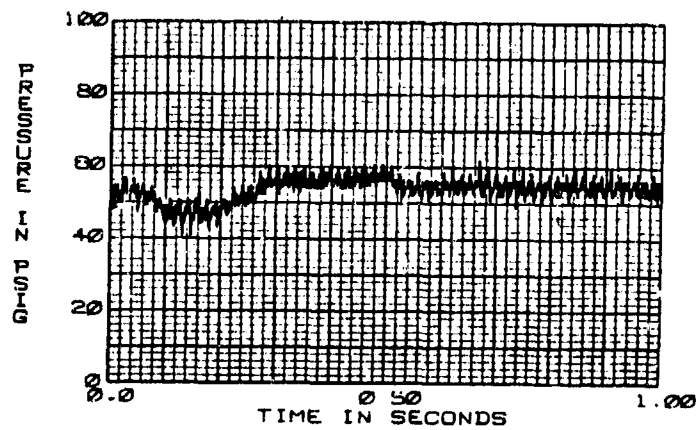


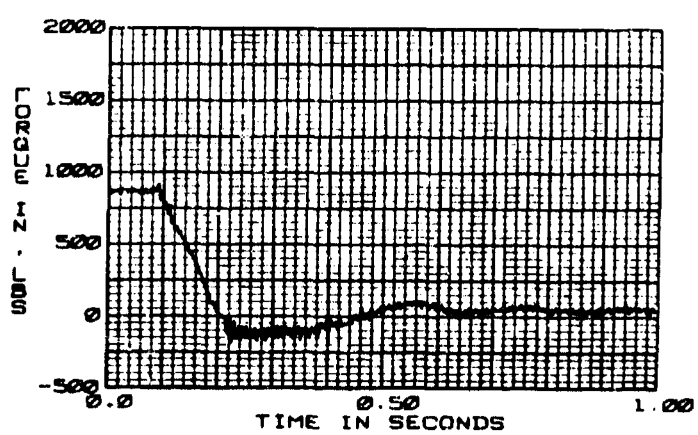
FIGURE 64 F-4 HYDRAULIC PUMP 105-5 TURN-OFF TRANSIENT, 104 CIS, 150°F



F-4 HYD PUMP
105-5-P5 TURN OFF TRANSIENT
104 CIS 150 F



F-4 HYD PUMP
105-5-P7 TURN OFF TRANSIENT
104 CIS 150 F



F-4 HYD PUMP
105-5-DT TURN OFF TRANSIENT
104 CIS 150 F

FIGURE 64 (CONTINUED)

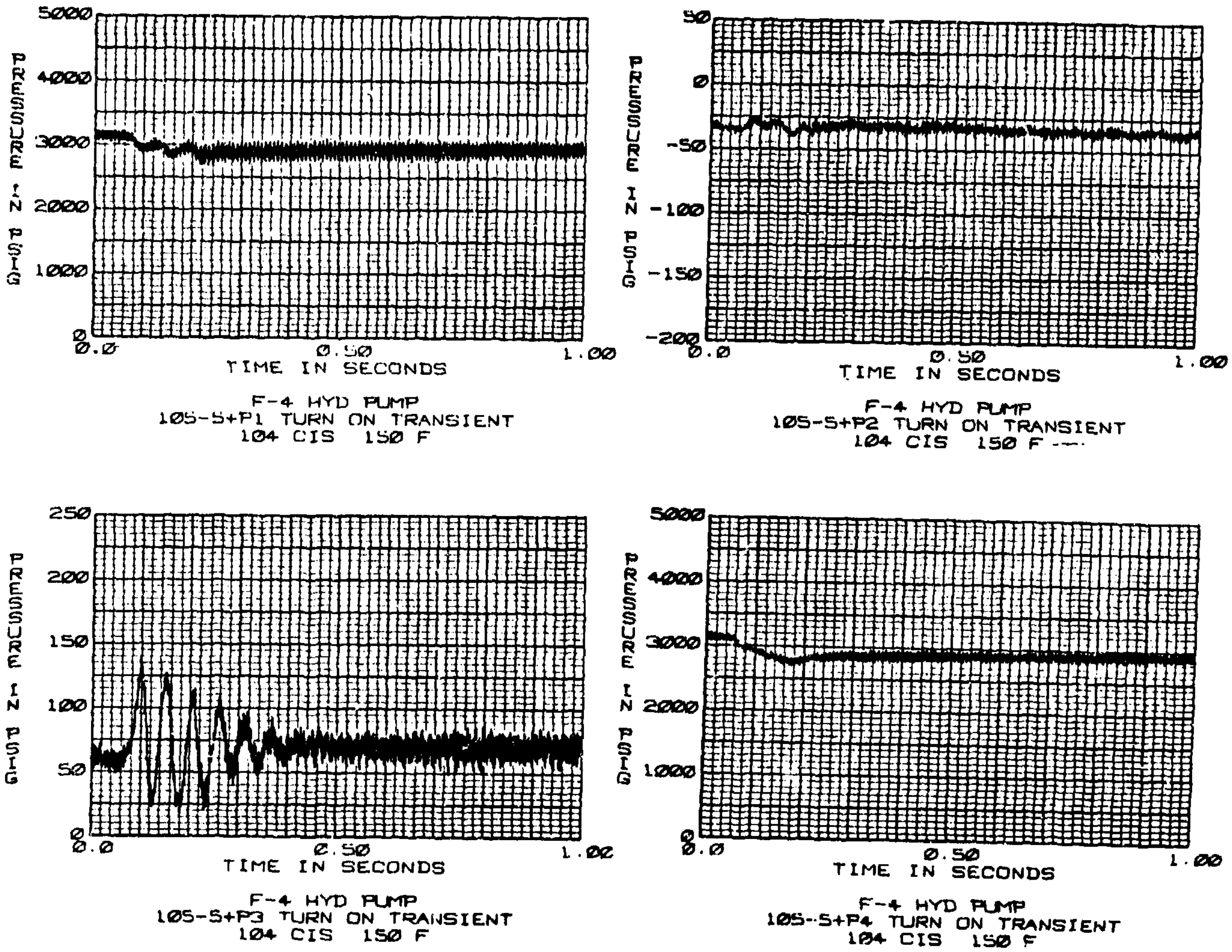
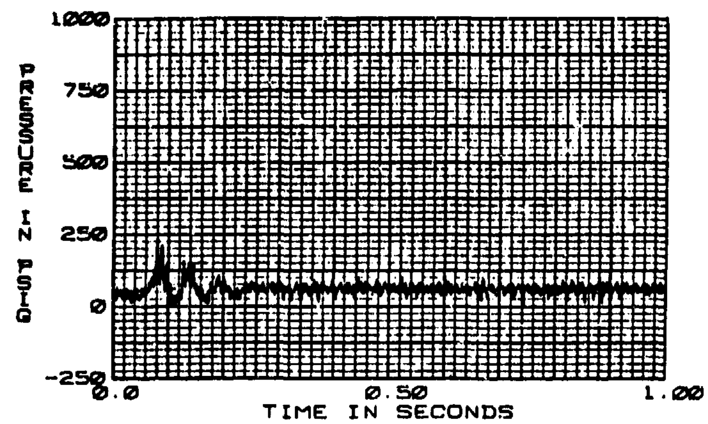
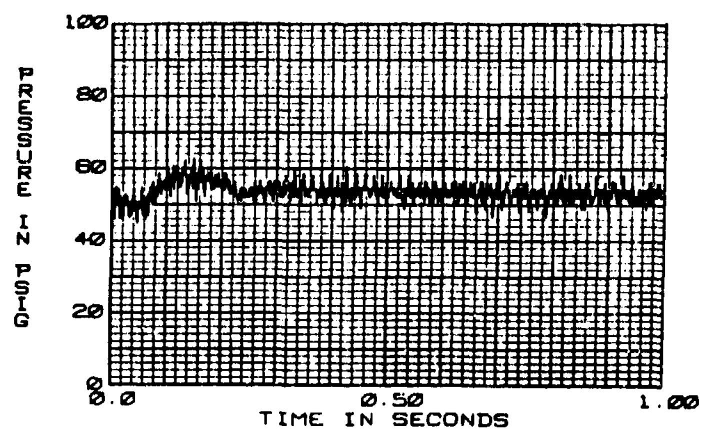


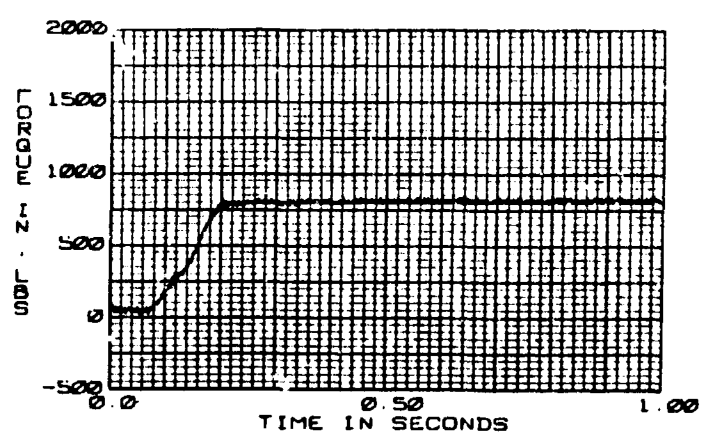
FIGURE 65 F-4 HYDRAULIC PUMP 105-5 TURN-ON TRANSIENT, 104 CIS, 150°F



F-4 HYD PUMP
105-5+P5 TURN ON TRANSIENT
104 CIS 150 F



F-4 HYD PUMP
105-5+P7 TURN ON TRANSIENT
104 CIS 150 F



F-4 HYD PUMP
105-5+DT TURN ON TRANSIENT
104 CIS 150 F

FIGURE 65 (CONTINUED)

c. Empirical Pump Model Usage and Correlation

The application of any empirical pump model depends on the amount of design and test data available on the pump/system to be simulated. An empirical pump model of a specific unit must be verified over a wide range of test conditions for it to be useful in other applications.

(1) First Order Pump Model

Data provided from previous pump testing was used during the initial development of the empirical pump models. The first order flow model gave good initial correlation with the test data.

The test system consisted of an F-15 hydraulic pump with a 1" x .058" wall x 393" long stainless steel line terminated by a transient control valve. The pump steady state input data was obtained from the test data presented in AFAPL-TR-77-63.

The following information is required input data for the first order pump flow model:

Pump Speed	4000 RPM
Pump Rated Speed	4600 RPM
Pump Flow @ Rated Speed	225 CIS
Pressure @ Zero Flow	2953. PSIA
Pressure @ Full Flow	2720. PSIA
First Order Time Constant	.04 SEC (turn-off)
Case Flow at Rated Flow and Pressure	4 CIS

The first order time constant was derived from the hanger response curve in Figure 66. The time for the hanger to oscillate one cycle is .064 seconds. During the time it is assumed that the hanger will reach 63.2% of its final value in .064 * .632 or .040 seconds. The exponential function reaches 63.2% of its final value in one time period so .040 is the first order time constant.

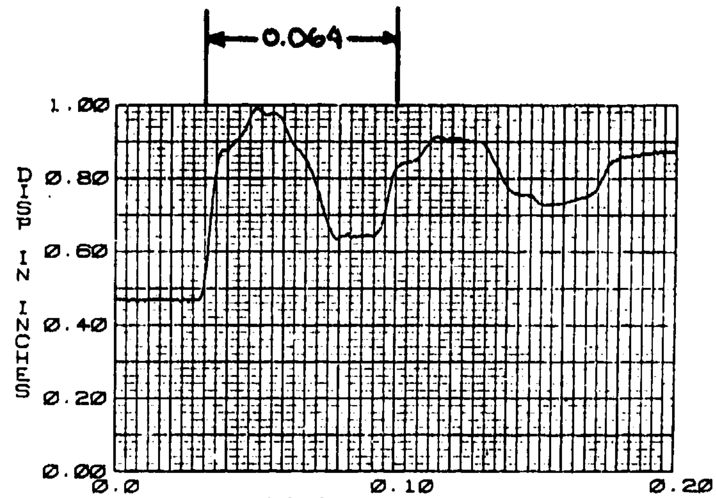


FIGURE 66 F-15 HYDRAULIC PUMP
65A03-XH TURN-OFF TRANSIENT
77 CIS 130 F

A HYTRAN system was assembled to simulate the turn-off transient at 77 CIS steady state flow and 130°F. The HYTRAN system schematic is in Figure 67. The HYTRAN input data file is in Table 3. The type 57 component is the empirical pump. The type 23 valve is identical to the type 21. The final steady state flow was 2.0 CIS. The HYTRAN computed pump pressure is shown in Figure 68. The test data for this run is plotted over the computer results. The computed results indicate good correlation with the test data. Superimposed on the computed pressure wave is a higher frequency pressure signal. The signal has a period of .015 seconds. This corresponds to the time required for the pressure signal to traverse the tube and return to its starting point. From Table 3 the velocity of sound in the line is 52325 in/sec and the line length is 393.25 in. The time for a pressure signal to leave the pump be reflected at the valve and return to the pump is

$$t = \frac{2\ell}{c} = \frac{2(393.25)}{52325} = .015 \text{ sec} \qquad (22)$$

The line model dynamic friction does not significantly dissipate the high frequency signal, and the pump model is incapable of absorbing the pressure energy.

To eliminate the pressure spikes a mathematical filter was added to the pump model. The time constant was set to .002 sec and the HYTRAN simulation was rerun. Plots of pump outlet pressure and flow are shown in Figures 69 and 70. The measured pump outlet pressure is overplotted with the computed results. The effects of a higher filter time constant (τ = .004) is shown in Figure 71 for a turn-off transient run. Selecting larger values can cause instability.

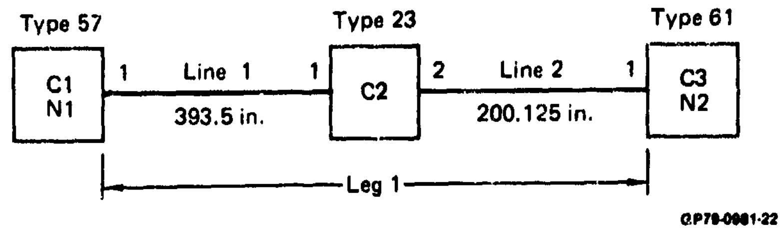


FIGURE 67 HYTRAN SCHEMATIC FOR THE EMPIRICAL PUMP MODEL

TABLE 3 FIRST ORDER EMPIRICAL PUMP MODEL INPUT DATA FOR TEST DATA RUN 65A03 TURN-OFF TRANSIENT

```
**** EMPIRICAL PUMP - FLOW MODEL (1ST ORDER) **** (DPUM57)

 THE TRANSIENT RESPONSE IS FROM T=0.0  TO T=   .200 SECONDS AT TIME INTERVALS OF DELT= .00020
          WITH OUTPUT POINTS PLOTTED AT INTERVALS OF   ,   .00200  SECONDS

FLUID DATA FOR  MIL-H-5606  AT  3000.0 PSIG. -    50.0 P.S.G AND 130.0 DEG F  IN   10.0 DEG F STEPS
                   VISCOSITY    -  .160E-01        .160E-01IN**2/SEC
                   DENSITY      -  .813E-04        .803E-04(LB-SEC**2)/IN**4
                   BULK MODULUS -  .223E+06        .187E+06PSI
                   VAPOUR PRESS.-  .200E+01 AT  130.0 DEG F

LINE DATA
LINE NO.      LENGTH        INTERNAL      WALL          MODULUS OF     BELK          CHARACTERISTIC VELOCITY OF
                            DIA           THICKNESS     ELASTICITY                   IMPEDANCE      SOUND
   1          393.5000      .8040         .0930         .300E+08       10.6204       6.9280         52325.4007
   2          200.1250      .9020         .0490         .300E+08       10.5320       6.0551         52325.4007
  COMP.    1 INTEGER DATA      1   57    1   -1    0    0    0    0    0    0    0    0    0    0    0    0
  REAL DATA CARD #    1     .4000E+04   .4600E+04   .2020E+03   .2053E+04   .2720E+04   .4000E-01   .4000E+01  0.
  COMP.    2 INTEGER DATA      2   23    3    1   -2    0    0    0    0    0    0    0    0    4    0    0
  REAL DATA CARD #    1     .2200E-01   .6500E+00  0.           0.          0.          0.          0.          0.
  REAL DATA CARD #    2    0.           .1000E-01   .2000E-01   .2000E+00  0.          0.          0.          0.
  REAL DATA CARD #    3     .6370E+00   .6370E+00   .1610E-01   .1600E-01  0.          0.          0.          0.
  COMP.    3 INTEGER DATA      3   61    1    2    0    0    0    0    0    0    0    0    0    0    0    0
  REAL DATA CARD #    1     .3000E+02  0.           0.          0.          0.          0.          0.          0.
     CPU TIME IN SECONDS =      2.313
```

Next a turn-on transient was tried in the same system. The initial steady state flow rate was 2.0 CIS at 130°F fluid temperature.

The first order time constant was determined from the hanger position in Figure 72 using a slightly larger time base.

$$T = .094 * .632 = .06 \text{ sec}$$

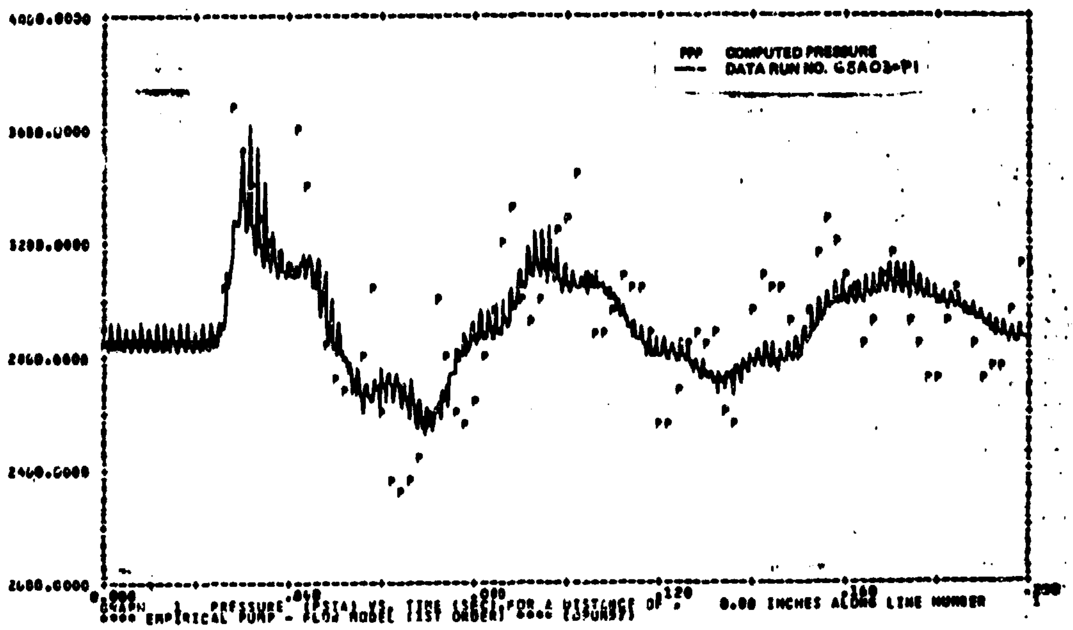


FIGURE 68 FIRST ORDER EMPIRICAL PUMP MODEL 65A03-P1 TRANSIENT SIMULATION

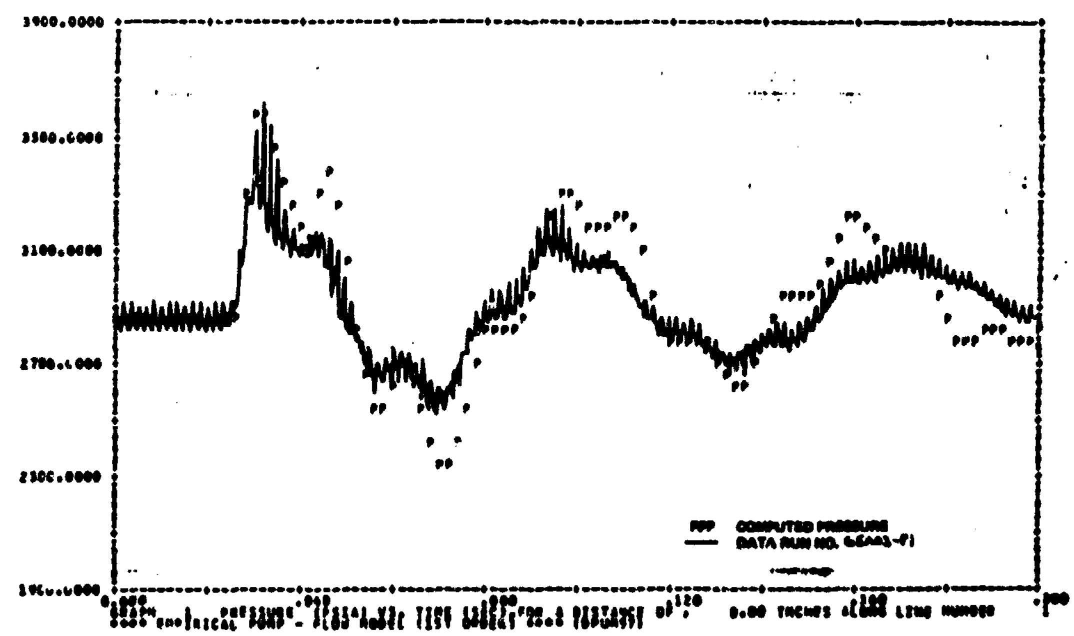


FIGURE 69 FIRST ORDER EMPIRICAL PUMP MODEL 65A03-P1 TRANSIENT SIMULATION WITH .002 FILTER TIME CONSTANT

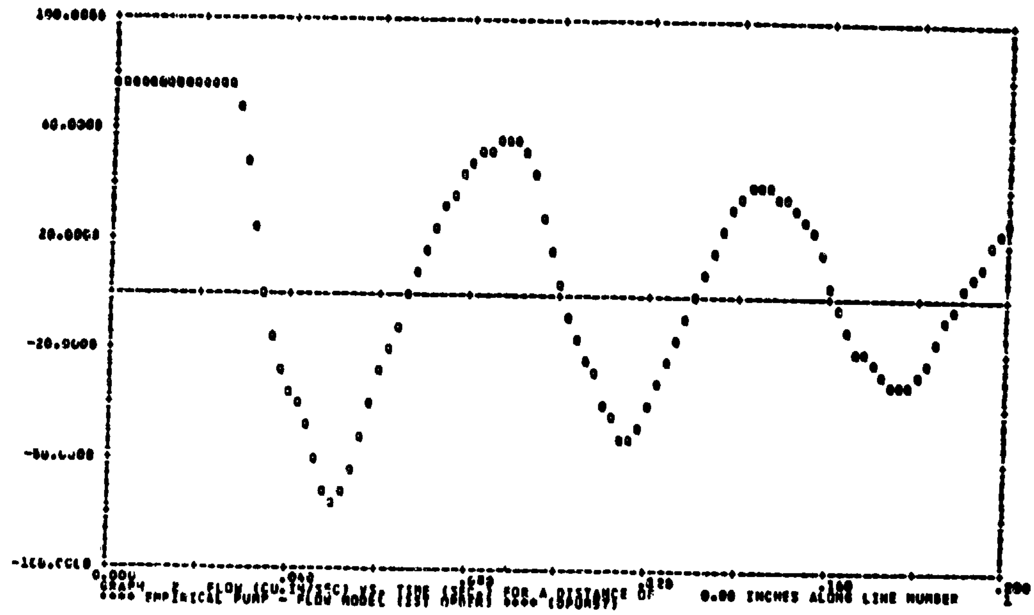


FIGURE 70 FIRST ORDER EMPIRICAL PUMP MODEL 65A03 PUMP OUTLET FLOW TRANSIENT SIMULATION

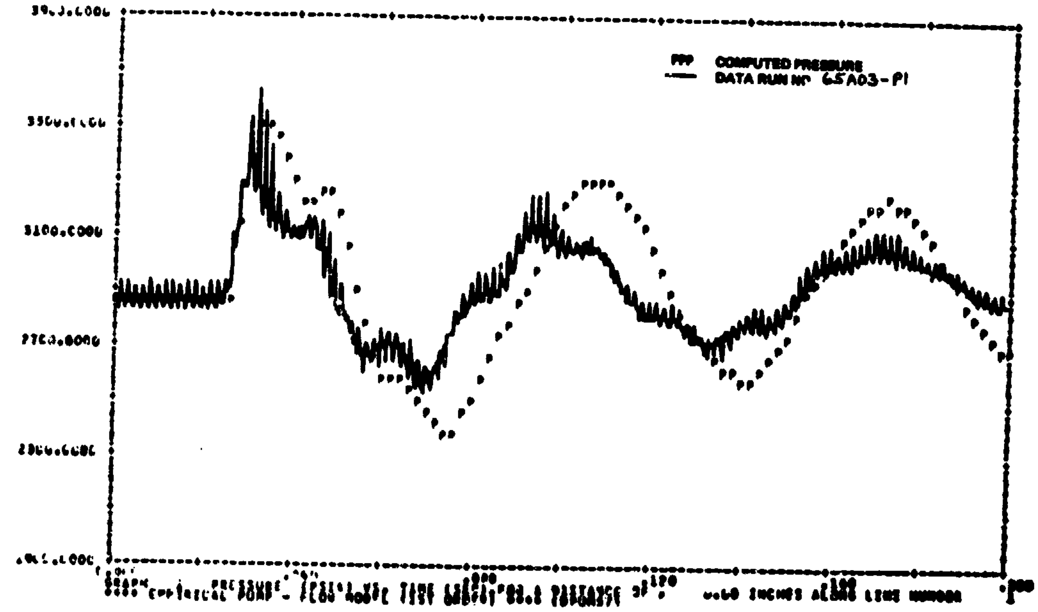


FIGURE 71 FIRST ORDER EMPIRICAL PUMP MODEL 65A03-P1 TRANSIENT SIMULATION WITH .004 FILTER TIME CONSTANT

Another method for obtaining the first time constant is by applying equation (9). From equation (9) the time constant equals RC. The hydraulic equivalent of resistance and capacitance can be expressed as

$$R = \frac{P}{Q} \tag{23}$$

$$C = \frac{Al}{\beta} \tag{24}$$

where

P = system pressure change across the load (psi)

Q = system flow change (CIS)

A = line flow area (in^2)

l = line length (in)

β = fluid bulk modulus (psi)

For the test system the pressure across the load went from 3000 psi to 100 psi. The flow rate was 77.0 CIS, and the fluid bulk modulus was 223000 psi. Substituting the appropriate values

$$R = \frac{3000-100}{77} = 38\ \frac{psi}{cis}$$

$$C = \frac{(.6137)(393)}{223000} = .001082\ \frac{in^3}{psi}$$

The system time constant is

$$T = 38\ \frac{psi}{cis} * .001082\ \frac{in^3}{psi} = .041\ sec$$

The time constant is larger at zero load flow, which means that the oscillation frequency and pressure gain will be lower.
The transient response is faster and more oscillatory for full flow to zero flow (turn-off transient) because of the higher gain and frequency response of the pump/system at rated pump flows.

A HYTRAN computer program turn-on transient was run using the .06 sec time constant. The results of the simulation shown in Figure 73 show a large error between the computed and measured data. A look at the pump inlet pressure in Figure 74 shows the pump cavitating transiently for approximately .03 seconds. The cavitation time was used to arrive at a new time constant for the simulation.

The HYTRAN input data for the turn-on transient run with the .03 sec time constant is in Table 4. The results of the simulation are shown on Figure 75, 76 and 77. Figure 76 is the pump outlet flow which resembles the hanger response.

Several HYTRAN computer runs were made with the empirical pump test data. Table 5 presents the HYTRAN input data for a turn-on transient with the F-15 hydraulic pump in a 280 in^3 system. The steady state input data was determined from figure 29. The first order pump model was used so the time constant was evaluated from Figure 78. The time period for a hanger cycle is .09 seconds. The time constant is .09 * .632 or .057 seconds. The computed outlet pressure and flow are shown in Figures 79 and 80. The pump outlet pressure data is overplotted on the HYTRAN output.

The transient valve closing rates were determined from the P4 and P5 (Figure 9) pressure transducers located on either side of the valve.

A turn-off transient run was simulated in the 280 in^3 system. The initial steady state flow was 77.0 CIS. The hanger response curve is shown in Figure 81. The HYTRAN data file is shown in Table 6. The computed pump outlet pressure and pressure at the control valve are overplotted with test data in Figures 82 and 83.

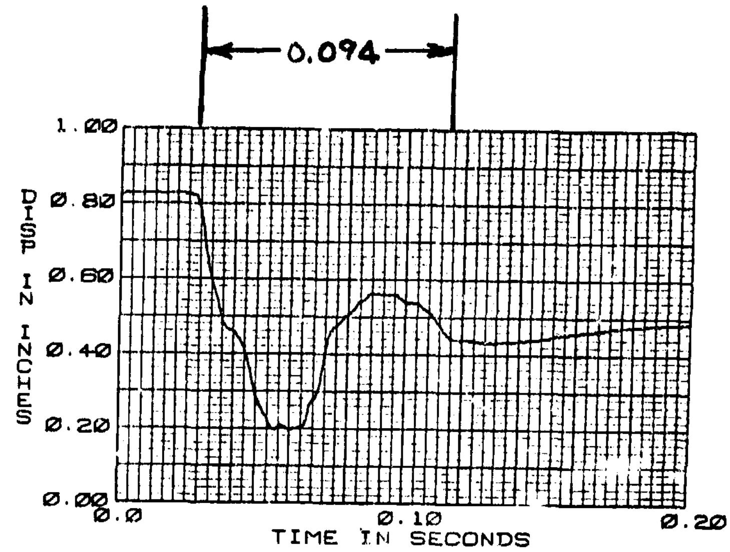


FIGURE 72 F-15 HYDRAULIC PUMP
65A03+XH TURN-ON TRANSIENT
77 CIS 130 F

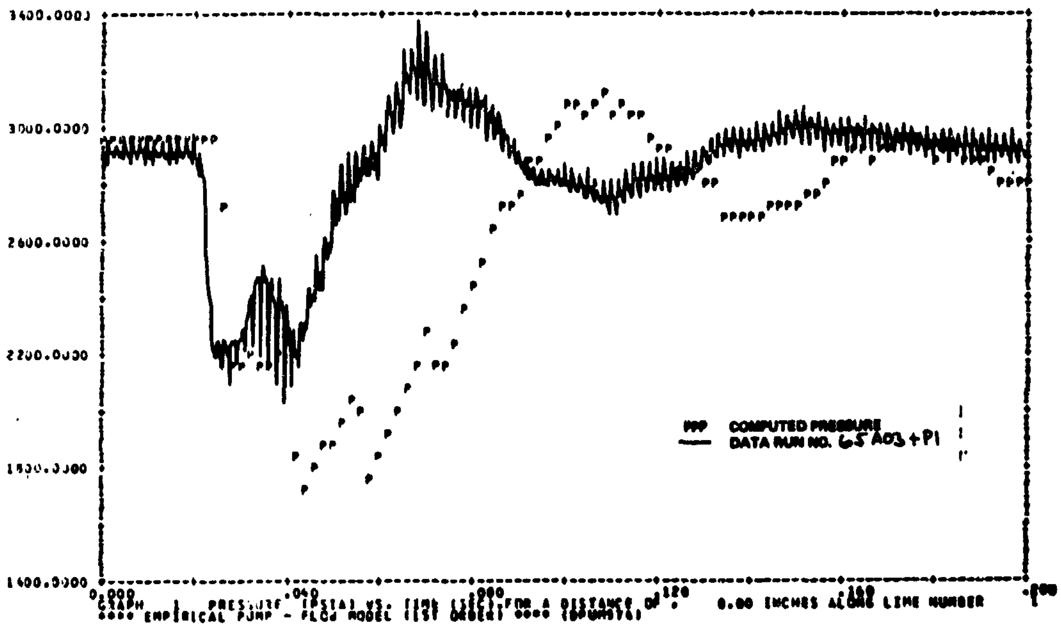


FIGURE 73 FIRST ORDER EMPIRICAL PUMP MODEL 65A03+P1
TRANSIENT/SIMULATION WITH .057 TIME CONSTANT

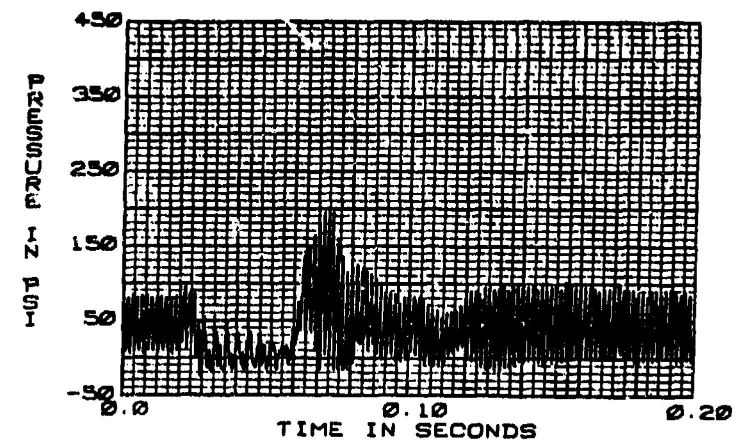


FIGURE 74 F-15 HYDRAULIC PUMP
65A03+PS TURN-ON TRANSIENT
77 CIS 130 F

TABLE 4 FIRST ORDER EMPIRICAL PUMP MODEL INPUT DATA FOR TEST DATA RUN 65A03 TURN-ON TRANSIENT

```
**** EMPIRICAL PUMP - FLOW MODEL (1ST ORDER) **** (DPUM97A)

THE TRANSIENT RESPONSE IS FROM T=0.0  TO T=   .200 SECONDS AT TIME INTERVALS OF DELT= .00020
      WITH OUTPUT POINTS PLOTTED AT INTERVALS OF   ,   .00200  SECONDS

FLUID DATA FOR  MIL-H-5606  AT  3000.0 PSIG, -    50.0 PSIG AND 130.0 DEG F  IN   10.0 DEG F STEPS
                   VISCOSITY    - .106E-01        .140E-01IN**2/SEC
                   DENSITY      - .913E-04        .803E-04(LB-SEC**2)/IN**4
                   BULK MODULUS - .229E+06        .107E+06PSI
                   VAPOUR PRESS.- .200E+01 AT  130.0 DEG F

LINE DATA
LINE NO.   LENGTH     INTERNAL   WALL       MODULUS OF   DELX      CHARACTERISTIC  VELOCITY OF
                      DIA        THICKNESS  ELASTICITY             IMPEDANCE       SOUND
   1       393.2500   .8840      .0500      .300E+08     10.6204   6.9208          52529.4097
   2       200.1250   .9920      .0490      .300E+08     10.9929   6.6501          52529.4097
 COMP.,   1 INTEGER DATA     1  57   7  -1   0   0   0   0   0   0   0   0   0   0   0   0
 REAL DATA CARD #   1    .4300E+04  .4600E+04  .2030E+03  .2050E+04  .2720E+04  .3000E-01  .4000E+01  0.
 COMP.,   2 INTEGER DATA     2  23   3   1  -2   0   0   0   0   1   0   0   0   4   0   0
 REAL DATA CARD #   1    .2200E-01  .6500E+00  0.         0.         0.         0.         0.         0.
 REAL DATA CARD #   2    0.         .1400E-01  .1600E-01  .2000E+00  0.         0.         0.         0.
 REAL DATA CARD #   3    .1630E-01  .1630E-01  .6370E+00  .6370E+00  0.         0.         0.         0.
 COMP.,   3 INTEGER DATA     3  61   1   2   0   0   0   0   0   0   0   3   0   0   0   0
 REAL DATA CARD #   1    .5900E+02  0.         0.         0.         0.         0.         0.         0.
       CPU TIME IN SECONDS =     2.749
```

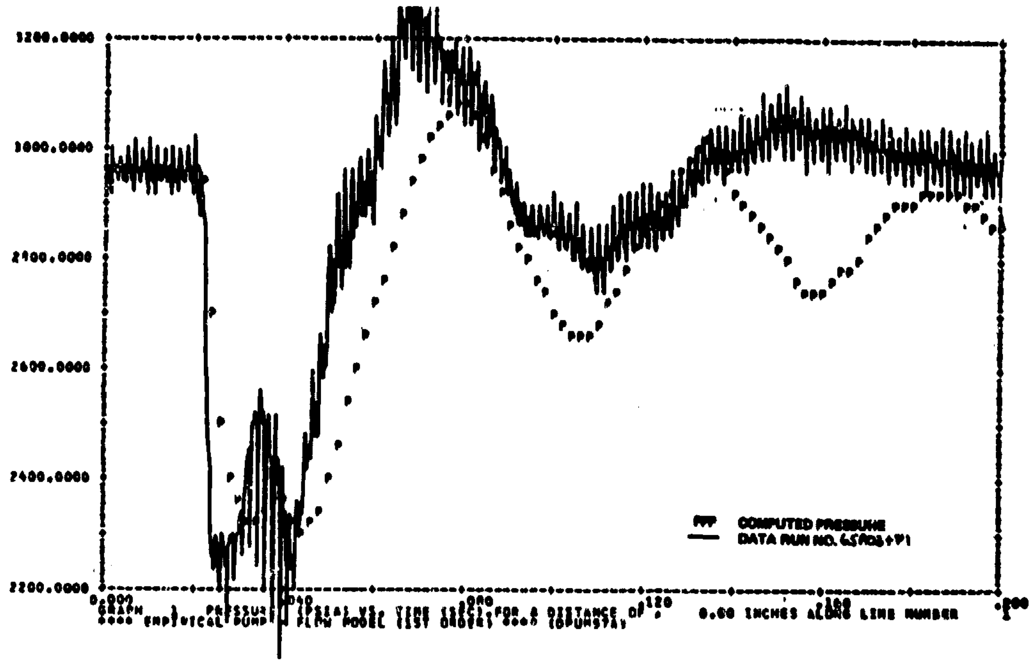


FIGURE 75 FIRST ORDER EMPIRICAL PUMP MODEL 65A03+P1
TRANSIENT/SIMULATION WITH .030 TIME CONSTANT

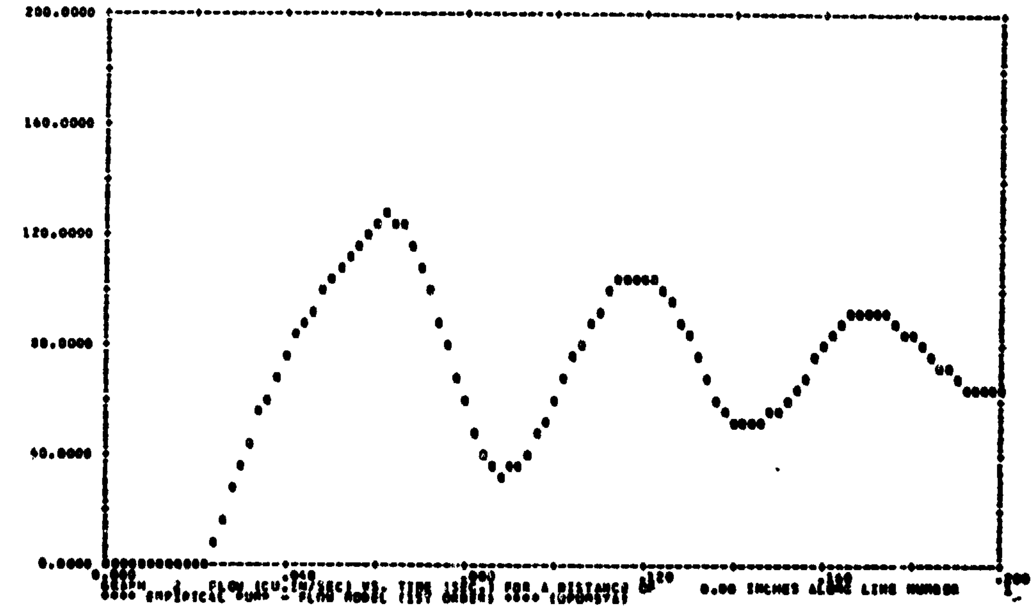


FIGURE 76 FIRST ORDER EMPIRICAL PUMP MODEL 65A03/PUMP OUTLET
FLOW TRANSIENT SIMULATION

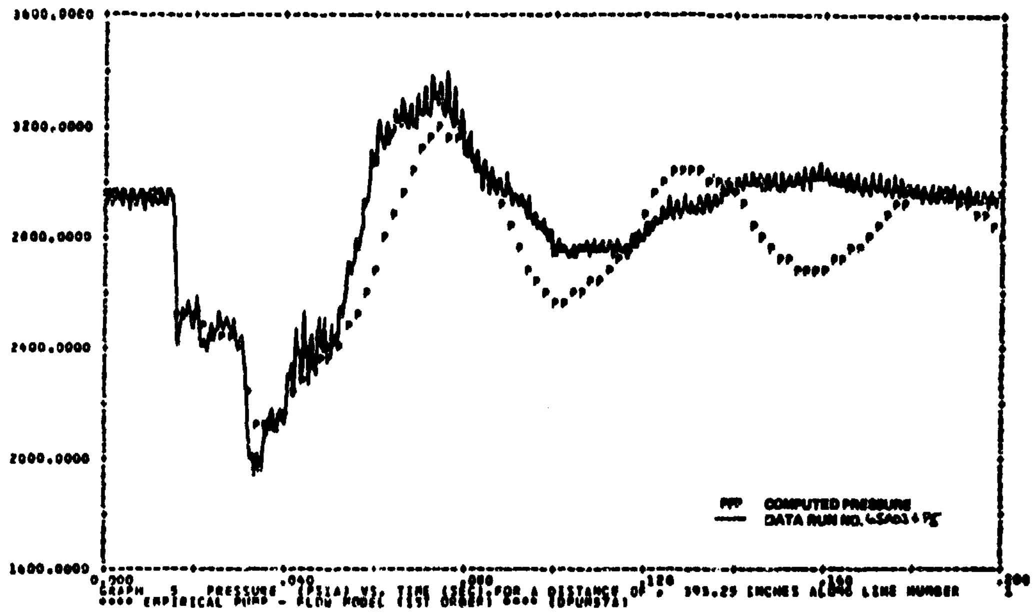


FIGURE 77 FIRST ORDER EMPIRICAL PUMP MODEL 65A03+P4
TRANSIENT SIMULATION

TABLE 5 FIRST ORDER EMPIRICAL PUMP MODEL INPUT DATA
FOR TEST DATA RUN 100-02 TURN-ON TRANSIENT

```
**** EMPIRICAL PUMP MODEL 280 IN**3 SYSTEM 100-02+XX ****(D1002P)
     .0005        .5      .005      150.
    2    3    1
    1                   2       360            452.        1.      .058    .3E+08
    2                                           40.        1.      .058    .3E+08
    1   57    1   -1
     3730.     4600.      225.     3165.     3100.      .057       8.8     .0001
    2   23    3    1   -2                                             4
      .022       .65
        0.      .050      .060        .5
      .017      .017     .4055     .4055
    3   61    1    2
      100.
    3    2
    1    1    2    3       10.
    1    1    0    1    2    1
    2    2    3    2       10.
    0    2    3    1
    1    5    0   00                   2
    1    4       .10      -.10      449.     -449.
    1    4    1    2    1   11    1   13    1   17
    1         0.     5000.    3        0.     5000.
```

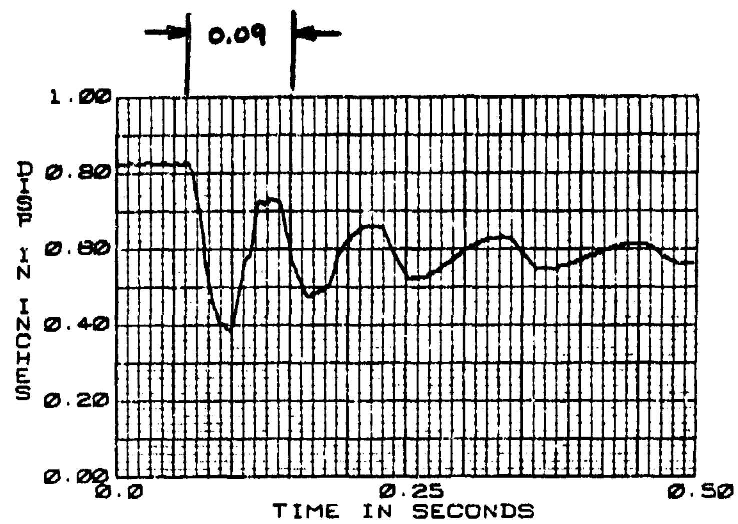


FIGURE 78 F-15 INSTRUMENTED PUMP
100-2+XH TURN-ON TRANSIENT
50 CIS 150 DEG F

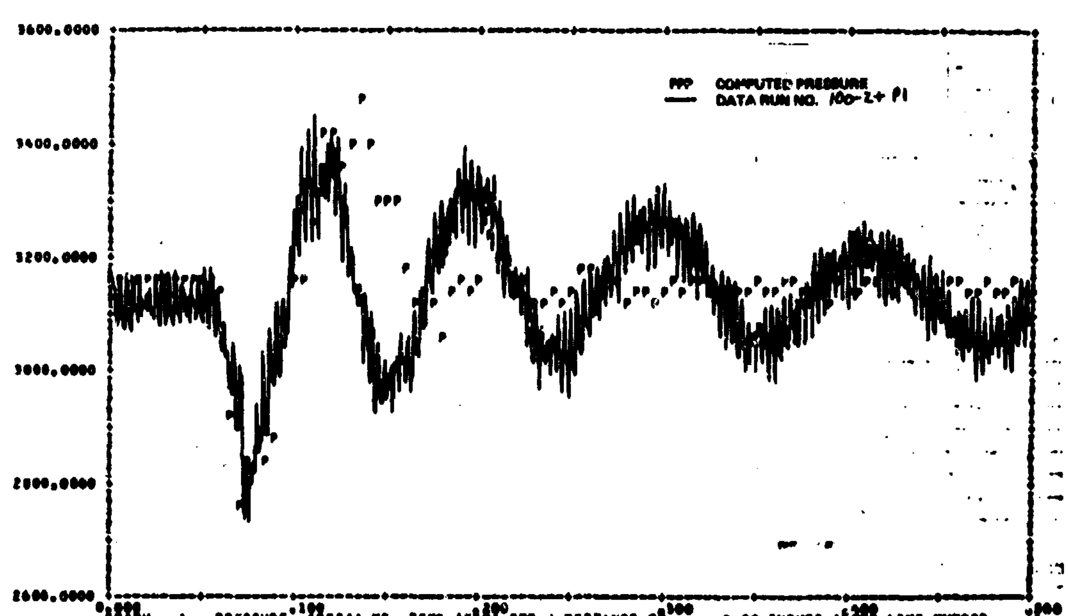


FIGURE 79 FIRST ORDER EMPIRICAL PUMP MODEL 100-2+P1
TRANSIENT SIMULATION

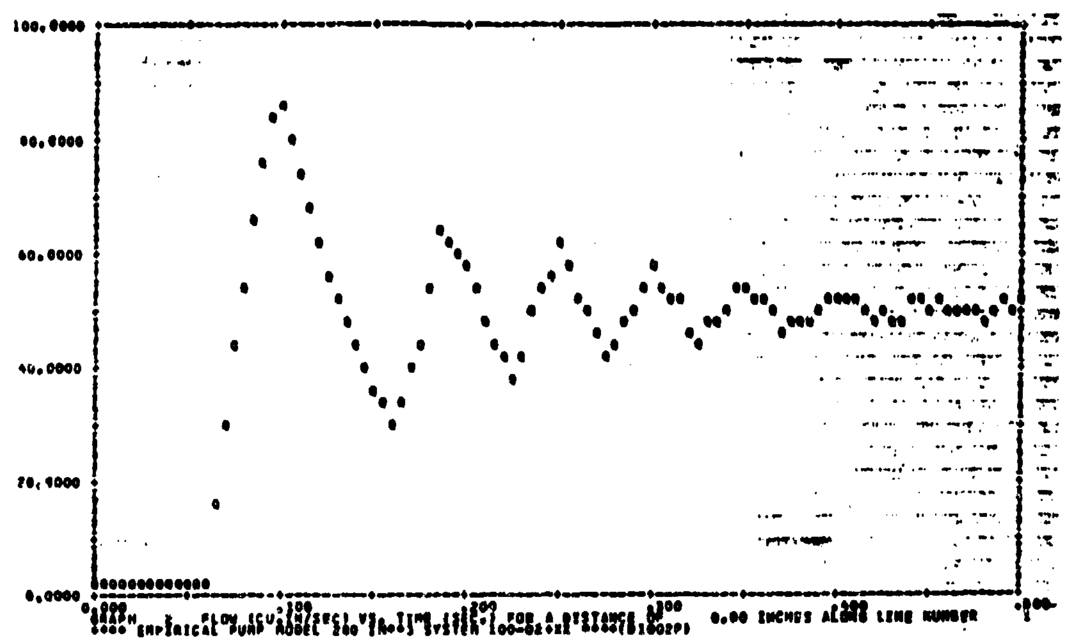


FIGURE 80 FIRST ORDER EMPIRICAL PUMP MODEL 100-2/PUMP OUTLET FLOW TRANSIENT SIMULATION

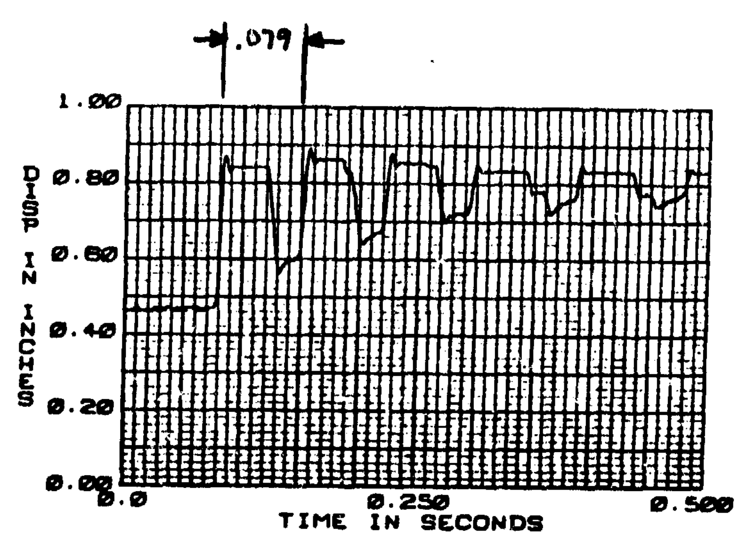


FIGURE 81 F-15 INSTRUMENTED PUMP 100-2-XH TURN-OFF TRANSIENT 30 CIS 150 DEG F

TABLE 6 FIRST ORDER EMPIRICAL PUMP MODEL INPUT DATA FOR TEST DATA RUN 100-02 TURN-OFF TRANSIENT

```
**** EMPIRICAL PUMP MODEL 280 IN**3 SYSTEM 100-02-XX ****(D1002M)
    .0005       .5      .005       150.
  2   3    1
  1                2         360             452.       1.       .058     .3E+08
  2                                           40.       1.       .058     .3E+08
  1  57    2  -1
   3750.     4600.      225.      3165.     3100.      .050       8.8        0.0
     2.0      .001       10.
  2  23    3   1   -2                                            4
    .022       .65
      0.      .078      .083         .5
   .6244     .6244      .017       .017
  3  61    1   2
    100.
  3   2
  1   1    2   3         10.
  1   1    0   1    2     1
  2   2    3   2         10.
  0   2    3   1
  1   5    1  00                          2
  1   4      .10       -.10       449.     -449.
  1   1    1   2    1     4    1     5    1    6
  1        0.    5000.    3         0.    5000.
```

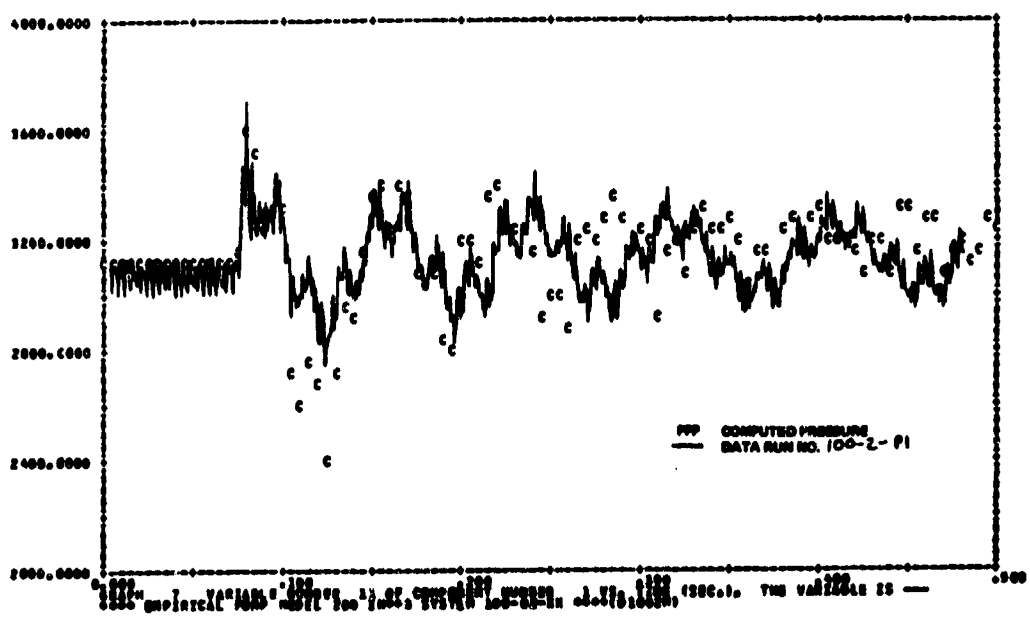


FIGURE 82 FIRST ORDER EMPIRICAL PUMP MODEL 100-2-P1 TRANSIENT SIMULATION

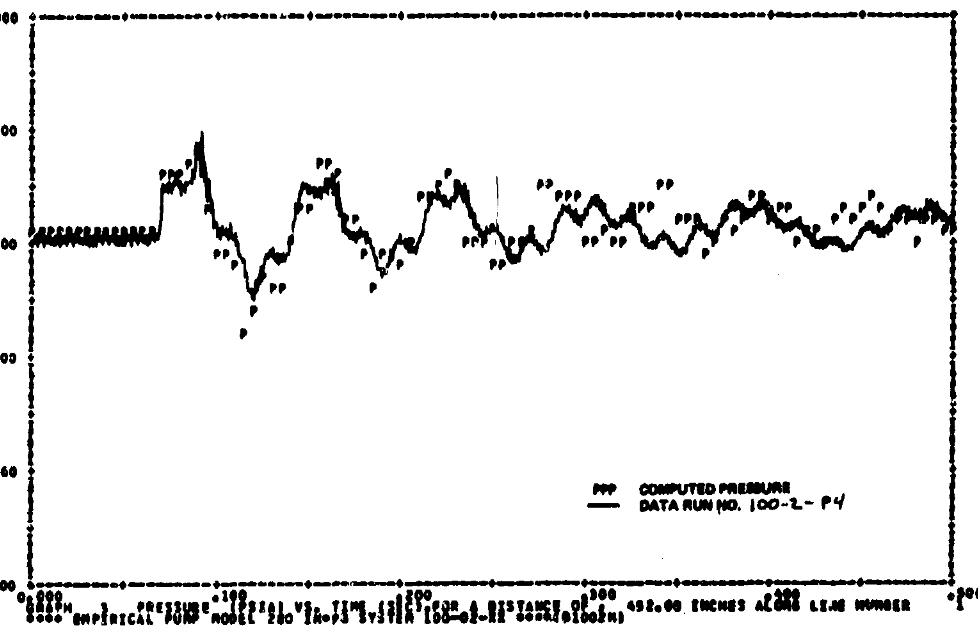


FIGURE 83 FIRST ORDER EMPIRICAL PUMP MODEL 100-2-P4
TRANSIENT SIMULATION

Table 7 presents the HYTRAN input data file for a turn-on transient in a 37 in^3 system. The initial steady state flow is 2.0 CIS. The transient control valve closing time was approximately .035 milliseconds. The first order time constant was estimated at .030 seconds. The pressures at the pump outlet and before the control valve are plotted over the computer results in Figures 84 and 85. The measured drive torque is shown in Figure 86 with the HYTRAN program predicted valve.

A turn-off transient in the same system was attempted. The damped natural frequency of the drive torque was approximately 25 HZ. The computed pump outlet pressure in Figure 87 continues to oscillate after the initial transient. The first order time constant is not large enough. Another run was made increasing the time constant on the mathematical filter used to dampen the pressure to .01 seconds. The computed outlet pressure in Figure 88 does not reach the peak pressure value of the measured data, but the simulation is stable. The HYTRAN input data is shown in Table 8.

The F-4 pump in a 538 in^3 system was simulated with the first order pump model. The HYTRAN input data is shown in Table 9. The first order time constant was estimated a .063 seconds. The predicted pressure output in Figures 89 and 91 correlate well with the test data. The computed pump outlet flow is in Figure 90.

The turn-off transient simulation input data is in Table 10. Since the undamped natural frequency of the drive torque was 6.172 Hz the time constant was set to .1024 seconds. Figures 92 and 93 show the simulated pump outlet pressure and flow. The computed pump outlet pressure matches the test data for the first transient spike in Figure 92. Then the computed data undershoots the desired results. The remainder of the simulation correlates well with the test data. Since there is no damping provided by the first order response the undershoot on the turn-off transient can be expected. A second order simulation could provide a much better response prediction. The turn-on transients do not indicate any type of undershoot or overshoot becuase the system natural frequency becomes larger when the valve is open.

TABLE 7 FIRST ORDER EMPIRICAL PUMP MODEL INPUT DATA FOR TEST DATA RUN 103-04 TURN-ON TRANSIENT

```
**** EMPIRICAL PUMP MODEL 37 IN**3 SYSTEM 103-04+XX ****(D1034P)
  .0005       .5      .005      100.
 2   3   1
 1               2       360            58.5       1.       .058    .3E+08
 2                                      40.        1.       .058    .3E+08
 1  57   1  -1
  1950.     4600.     225.      3165.     3100.     .030      8.8     .0001
 2  23   3   1  -2                                            4
  .022       .65
   0.       .050      .085       .5
  .017      .017     .7756     .7756
 3  61   1   2
  100.
 3   2
 1   1   2   3        10.
 1   1   0   1   2     1
 2   2   3   2        10.
 0   2   3   1
 1   5   1  00                          2
 1   4      .01      -.01       58.0     -58.0
 1   4   1   2   1    11     1   13     1   17
 1        0.     5000.   3       0.     5000.
```

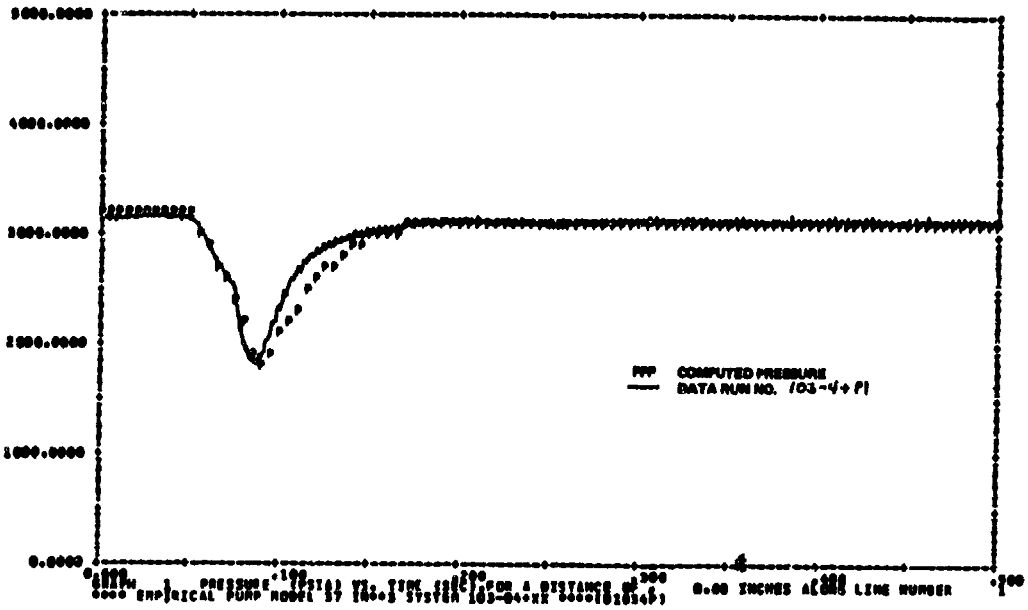


FIGURE 84 FIRST ORDER EMPIRICAL PUMP MODEL 103-4+P1 TRANSIENT SIMULATION

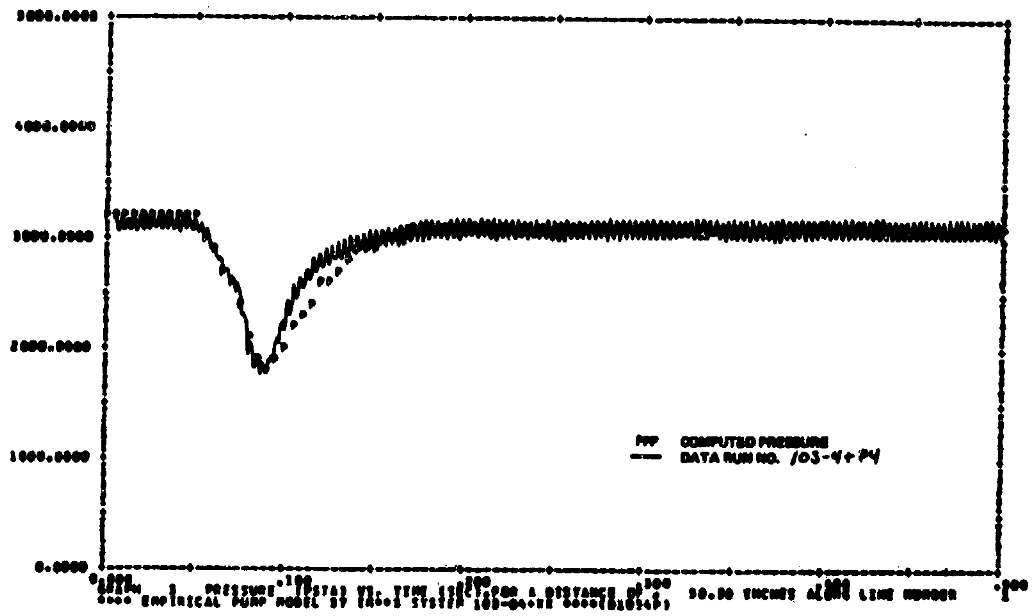


FIGURE 85 FIRST ORDER EMPIRICAL PUMP MODEL 103-4+P4 TRANSIENT SIMULATION

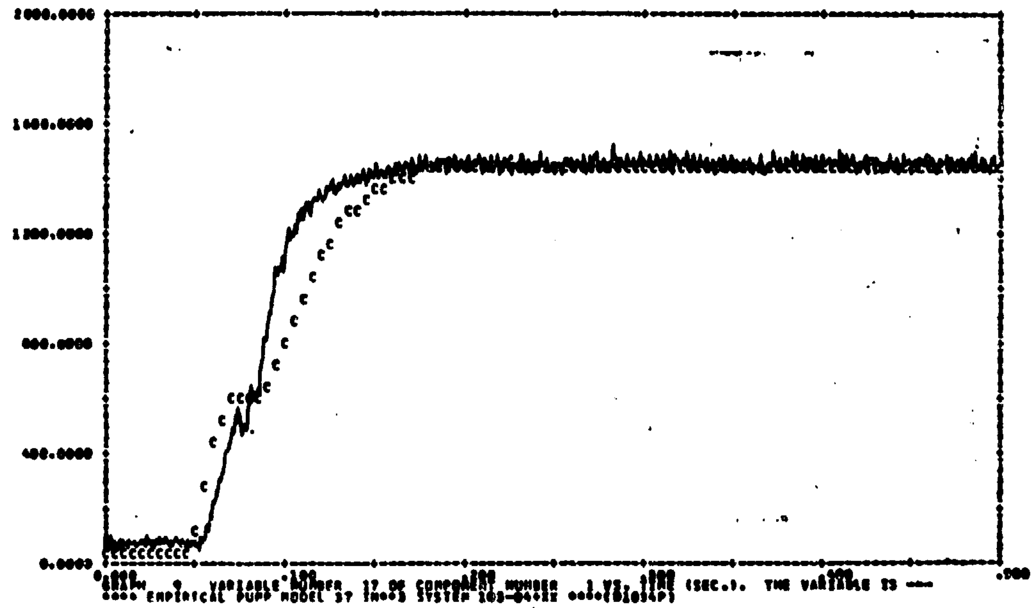


FIGURE 86 FIRST ORDER EMPIRICAL PUMP MODEL 103-4+DT TRANSIENT SIMULATION

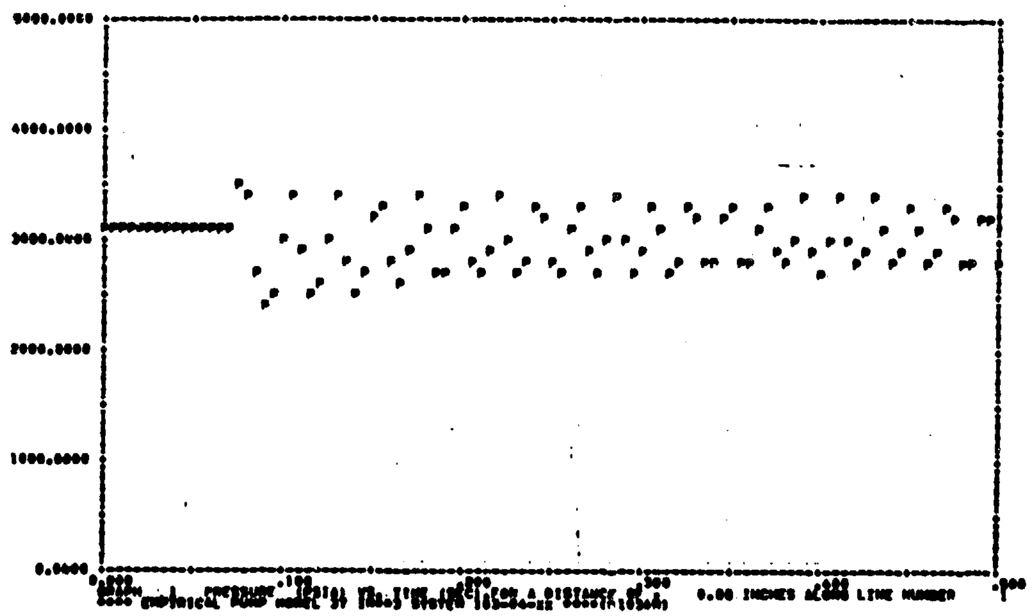


FIGURE 87 FIRST ORDER EMPIRICAL PUMP MODEL 103-4-P1 TRANSIENT SIMULATION

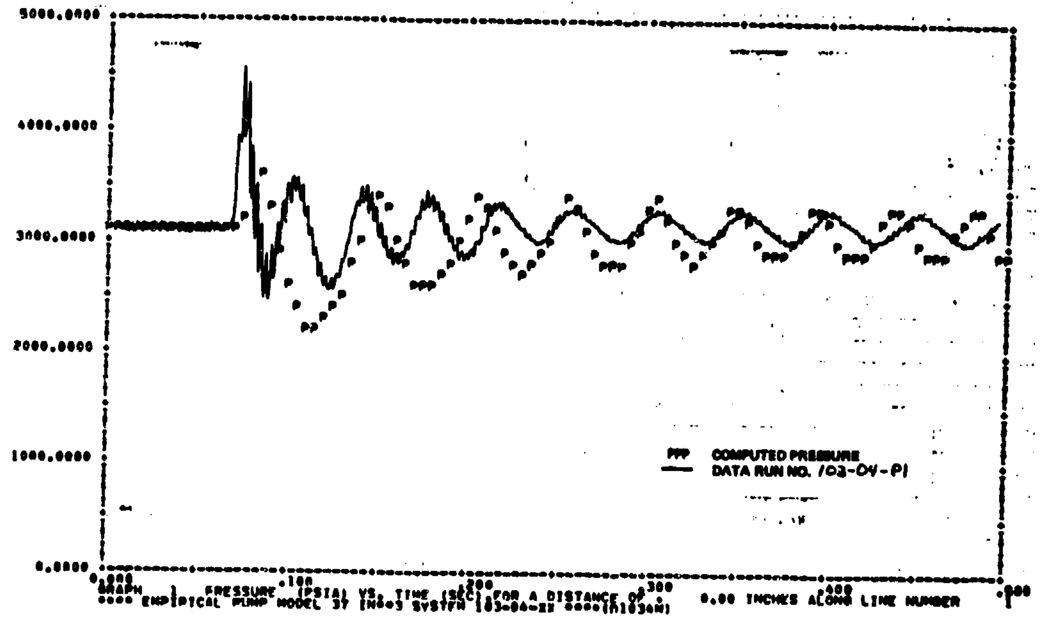


FIGURE 88 FIRST ORDER EMPIRICAL PUMP MODEL 103-4-P1
TRANSIENT SIMULATION WITH .01 SECOND FILTER TIME CONSTANT

TABLE 8 FIRST ORDER EMPIRICAL PUMP MODEL INPUT DATA FOR TEST DATA RUN 103-04 TURN-OFF TRANSIENT

```
**** EMPIRICAL PUMP MODEL 37 IN**3 SYSTEM 103-04-XX ****(D1034M)
   .00025        .5       .005      100.
  2    3    1
  1                  2        360           58.5        1.       .058     .3E+08
  2                                          40.        1.       .058     .3E+08
  1   57    1   -1
   1950.     4600.      225.     3165.     3100.      .032       8.8      .0100
  2   23    3    1   -2                                          4
    .022       .65
      0.      .070      .080        .5
   .7756     .7756      .017      .017
  3   61    1    2
    100.
  3    2
  1    1    2    3        10.
  1    1    0    1    2    1
  2    2    3    2        10.
  0    2    3    1
  1    5    0   00                          2
  1    4       .01      -.01      58.0     -58.0
  1    4    1    2    1   11    1   13    1   17
  1         0.      5000.    3        0.    5000.
```

TABLE 9 FIRST ORDER EMPIRICAL PUMP MODEL INPUT DATA FOR TEST DATA RUN 105-4 TURN-ON TRANSIENT

```
**** EMPIRICAL PUMP MODEL 538 IN**3 SYSTEM 105-04+XX ****(D1054P)
    .001         1.       .01       148.
  2    3    1
  1                  2        360         876.75        1.       .058     .3E+08
  2                                          40.        1.       .058     .3E+08
  1   57    1   -1
   3750.     3750.      106.     3165.     3100.      .063       3.8      .0001
  2   23    3    1   -2                                          4
    .022       .65
      0.      .050      .085        1.
    .017      .017     .8434     .8434
  3   61    1    2
    100.
  3    2
  1    1    2    3        10.
  1    1    0    1    2    1
  2    2    3    2        10.
  0    2    3    1
  1    5    1   00                          2
  1    4       .01      -.01      876.     -876.
  1    4    1    2    1   11    1   13    1   17
  1         0.      500C.    3        0.    5000.
```

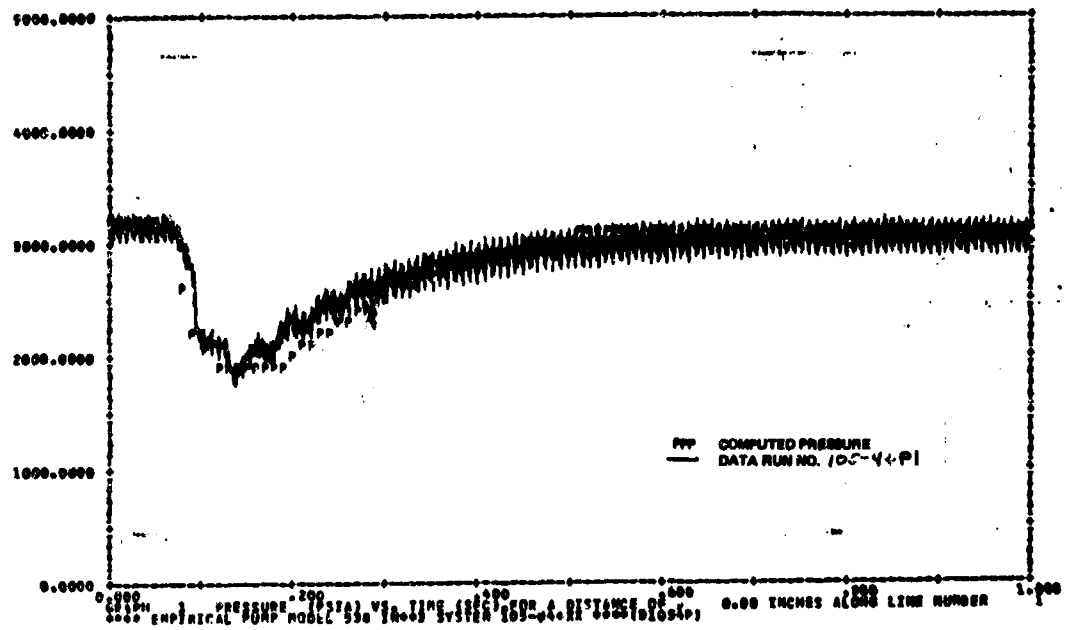


FIGURE 89 FIRST ORDER EMPIRICAL PUMP MODEL 105-4+P1
TRANSIENT SIMULATION

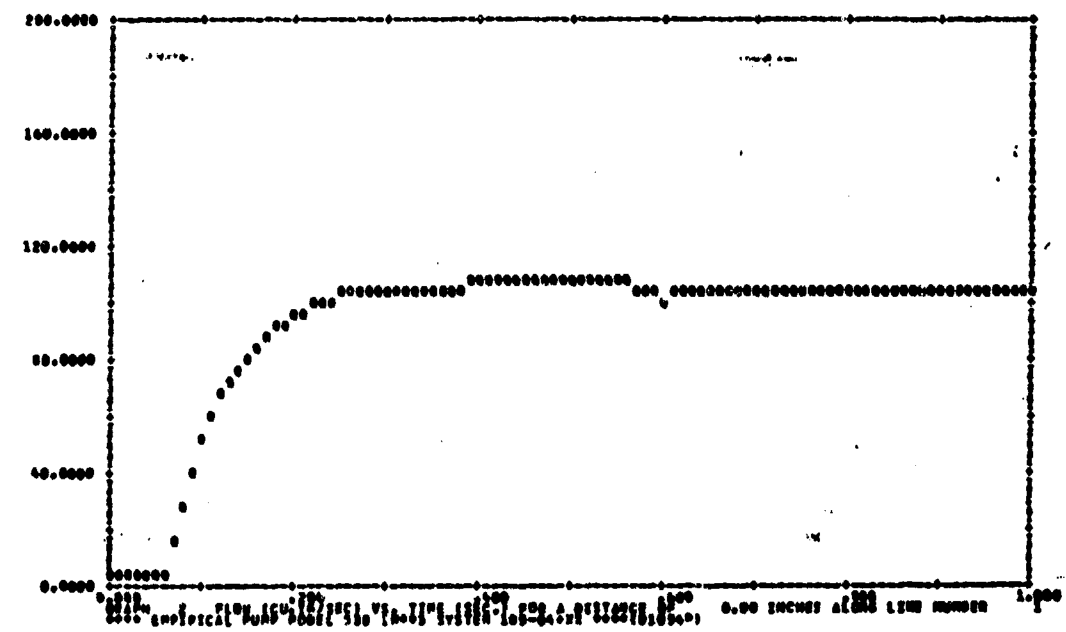


FIGURE 90 FIRST ORDER EMPIRICAL PUMP MODEL 105-4
PUMP OUTLET FLOW TURN-ON TRANSIENT SIMULATION

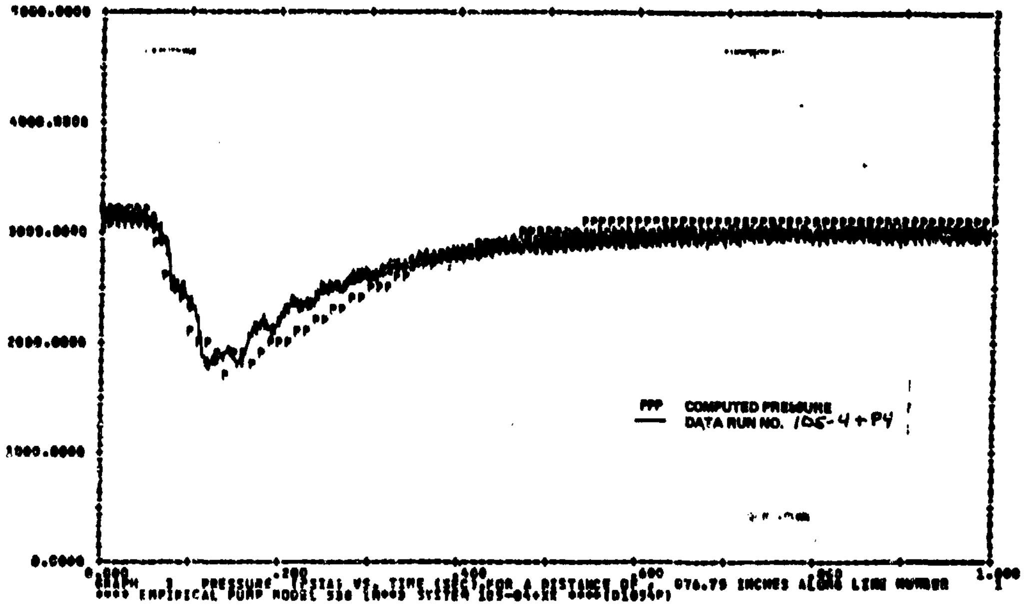


FIGURE 91 FIRST ORDER EMPIRICAL PUMP MODEL 105-4+P4 TRANSIENT SIMULATION

TABLE 10 FIRST ORDER EMPIRICAL PUMP MODEL INPUT DATA FOR TEST DATA RUN 105-4 TURN-OFF TRANSIENT

```
**** EMPIRICAL PUMP MODEL 538 IN**3 SYSTEM 105-04-XX ****(D1054M)
   .0005       1.        .01       145.
  2    3    1
  1                  2        360          876.75        1.       .058     .3E+08
  2                                           40.        1.       .058     .3E+08
  1   .57   1   -1
   3750.     3750.      106.      3165.     3100.       .102       3.8      .0001
  2   23    3    1   -2                                            4
    .022       .65
      0.      .070      .080        1.
   .8434     .8434      .017      .017
  3   61    1    2
    100.
  3    2
  1    1    2    3        10.
  1    1    0    1    2    1
  2    2    3    2        10.
  0    2    3    1
  1    5    1   00                          2
  1    4       .01       -.01       876.     -876.
  1    4    1    2    1   11    1   13    1   17
  1         0.      5000.   3         0.     5000.
```

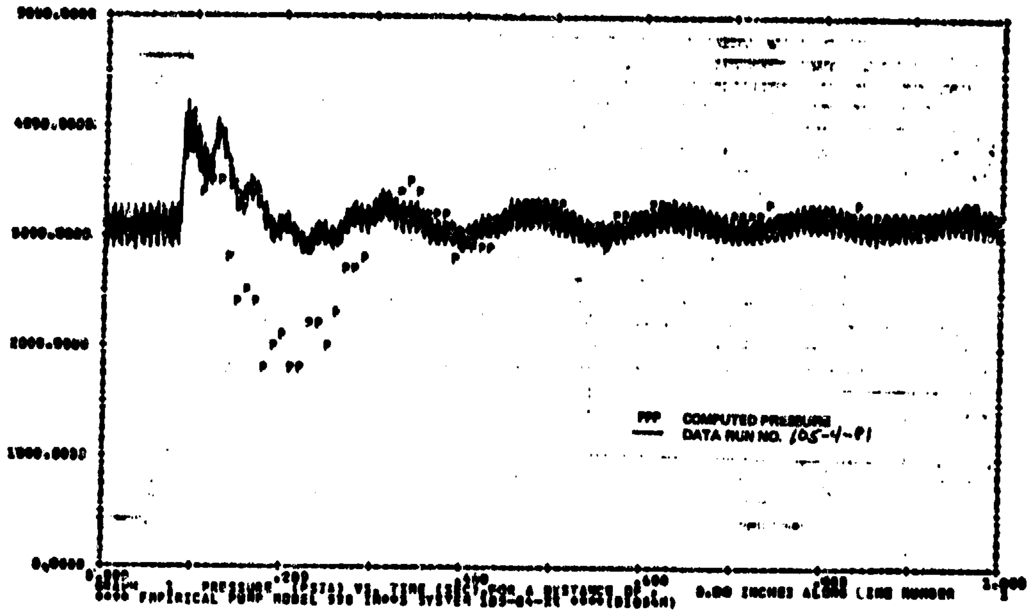


FIGURE 92 FIRST ORDER EMPIRICAL PUMP MODEL 105-4-P1
TRANSIENT SIMULATION

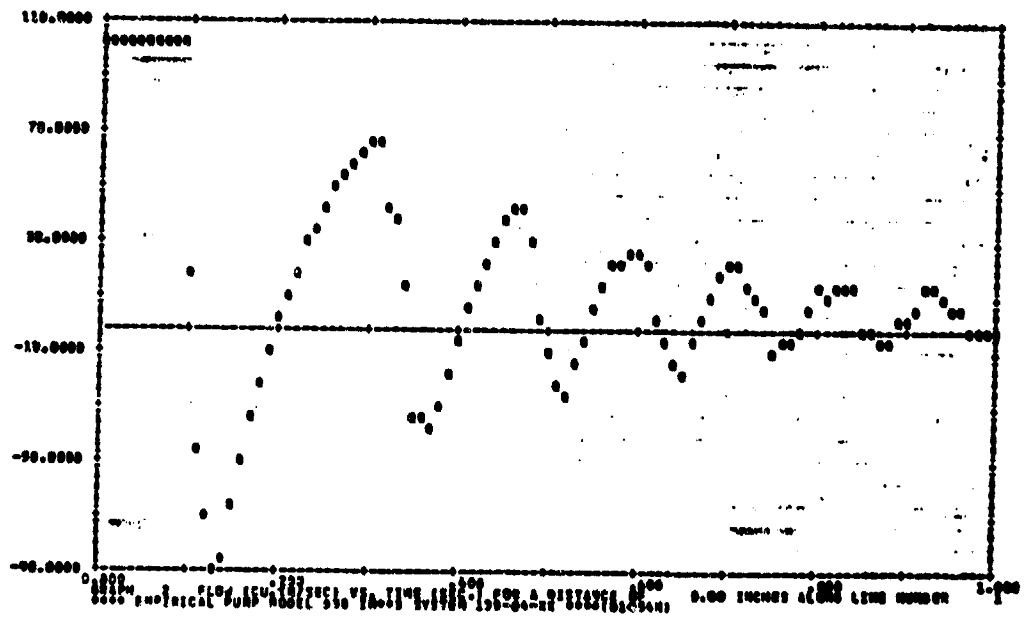


FIGURE 93 FIRST ORDER EMPIRICAL PUMP MODEL 105-4 PUMP OUTLET
FLOW TURN-OFF TRANSIENT SIMULATION

The F-15 pump was simulated by the first order model in a 546 in^3 system. The HYTRAN input data in Table 11 was used to predict the computed output pressures and flow in Figures 94, 95 and 96. The first order time constant was estimated as .030 seconds using the rise time in the drive torque data in Figure 97.

The turn-off simulation using the input data of Table 12 and the time constant from Figure 98 also shows the undershoot characteristics in Figures 99, 100 and 101.

(2) Second Order Model

The first order response will provide a good approximation of the pump characteristics. In short systems with fast transient control valves the pressure oscillation frequency can be significantly higher. A second order function can simulate the pump response.

The second order pump flow model requires more input data than the first order model, but the data is readily available from the test results. The following data is required for the second order pump model for a turn-off transient in a 37 in^3 system.

Pump Speed	1950 RPM
Pump Rated Speed	4600 RPM
Pump Flow @ Rated Speed	225 CIS
Pressure @ Zero Flow	3125 PSIA
Pressure @ Full Flow	3100 PSIA
Natural Frequency	150 RAD/SEC
Case Flow @ Rated Flow and Pressure	5 CIS
Damping Factor	.05
Final Steady State Flow	2 CIS
Flow Response Delay	.004 SEC
Pump and Pump Manifold Volume	10. IN^3

TABLE 11 FIRST ORDER EMPIRICAL PUMP MODEL INPUT DATA
FOR TEST DATA RUN 102-03 TURN-OFF TRANSIENT

```
**** EMPIRICAL PUMP MODEL 546 IN**3 SYSTEM 102-03+XX ****(D1023P)
    .0005        .5       .005       150.
  2    3    1
  1                  2         360            887.75       1.       .058    .3E+08
  2                                              40.       1.       .058    .3E+08
  1   57    1   -1
   3750.     4600.       225.      3165.      3100.       .030       8.8     .0001
  2   23    3    1   -2                                               4
    .022         .65
      0.        .050       .085         .5
    .017        .017      .9489      .9489
  3   61    1    2
    100.
  3    2
  1    1    2    3       10.
  1    1    0    1    2    1
  2    2    3    2       10.
  0    2    3    1
  1    5    0   00                         2
  1    4        .01      -.01       887.    -887.
  1    4    1    2    1   11    1   13   1   17
  1         0.      5000.   3          0.   5000.
```

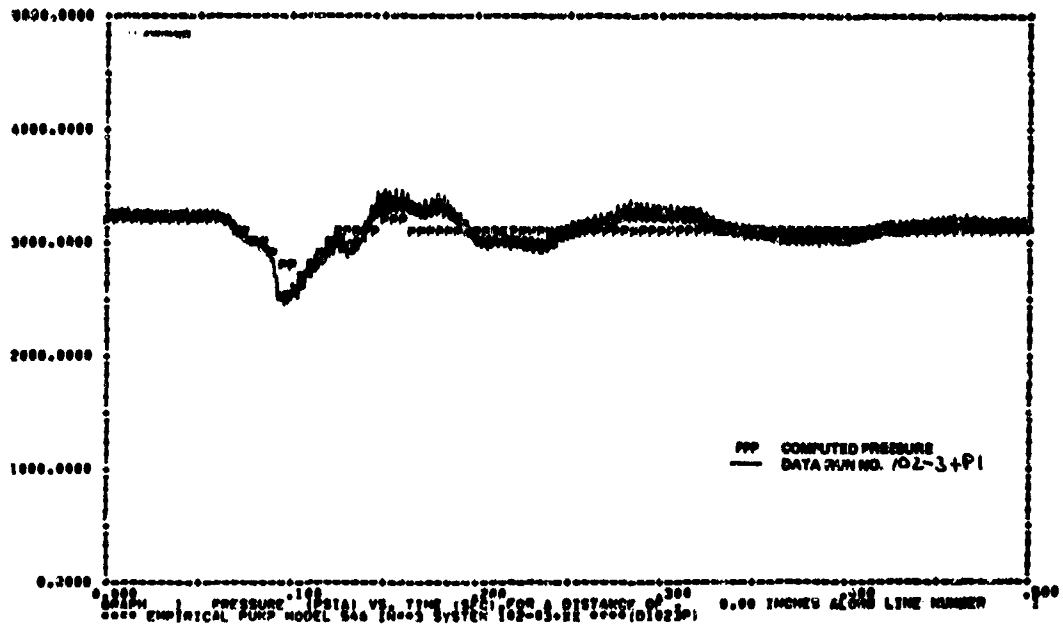


FIGURE 94 FIRST ORDER EMPIRICAL PUMP MODEL 102-3+P1 TRANSIENT SIMULATION

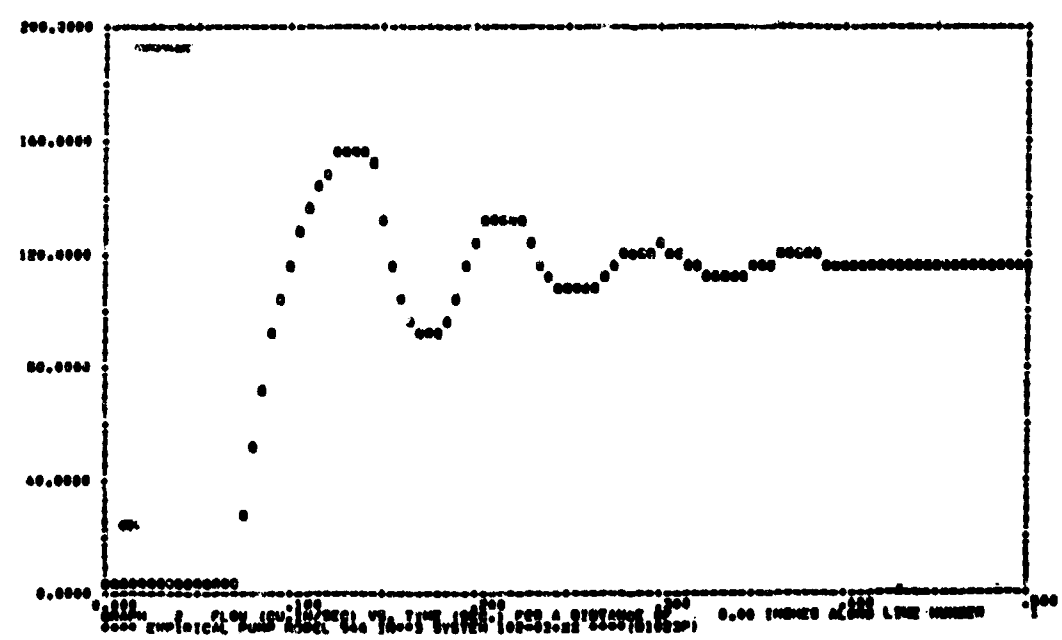


FIGURE 95 FIRST ORDER EMPIRICAL PUMP MODEL 102-3 PUMP OUTLET FLOW TURN-ON TRANSIENT SIMULATION

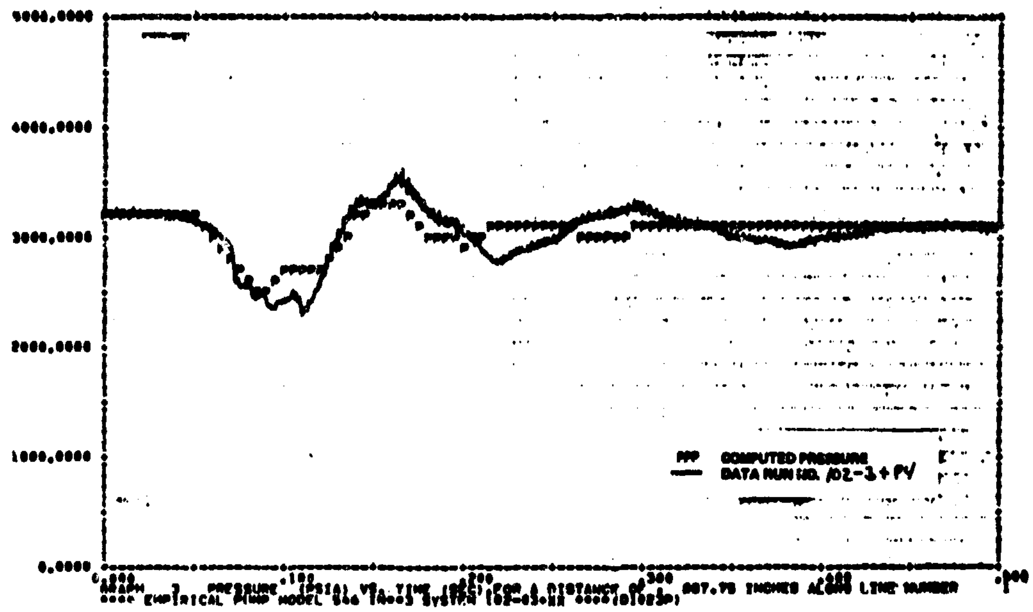


FIGURE 96 FIRST ORDER EMPIRICAL PUMP MODEL 102-3+P4 TRANSIENT SIMULATION

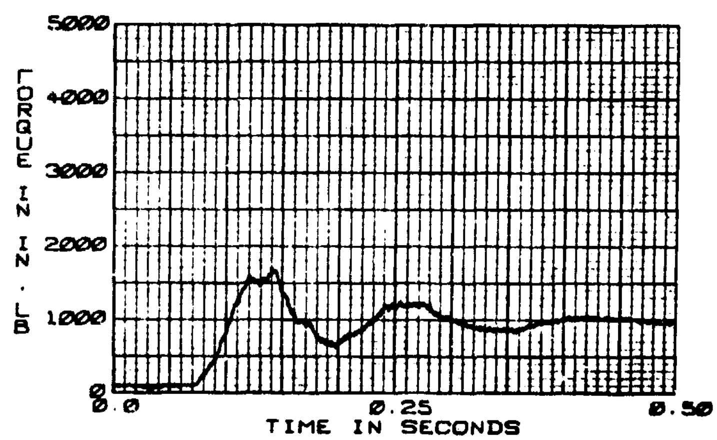


FIGURE 97 F-15 HYD PUMP
102-3+DT TURN ON TRANSIENT
117 CIS 155 F

TABLE 12 FIRST ORDER EMPIRICAL PUMP MODEL INPUT DATA FOR TEST DATA RUN 102-03 TURN-OFF TRANSIENT

```
**** EMPIRICAL PUMP MODEL 546 IN**3 SYSTEM 102-03-XX ****(D1023H)
    .0005        .5       .005       150.
  2    3    1
  1                  2         360          887.75       1.       .058    .3E+08
  2                                            40.       1.       .058    .3E+08
  1   57    1   -1
   3750.     4600.       225.      3165.      3100.     .079       8.8     .0001
  2   23    3    1   -2                                            4
    .022       .65
      0.      .060       .070         .5
   .9489     .9489       .017       .017
  3   61    1    2
    100.
  3    2
  1    1    2    3         10.
  4    1    0    1    2      1
  2    2    3    2         10.
  0    2    3    1
  1    5    0   00                        2
  1    4       .01       -.01       887.     -887.
  1    4    1    2    1     11    1   13    1   17
  1         0.      5000.    3          0.     5000.
```

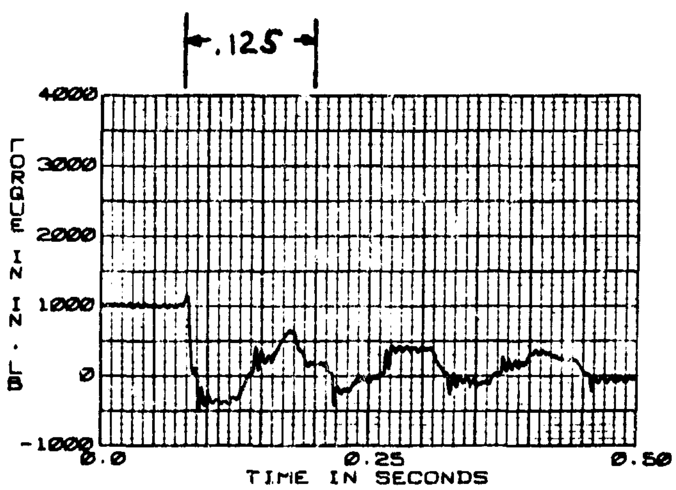


FIGURE 98 F-15 HYD PUMP
102-3-DT TURN-OFF TRANSIENT
117 CIS 150 F

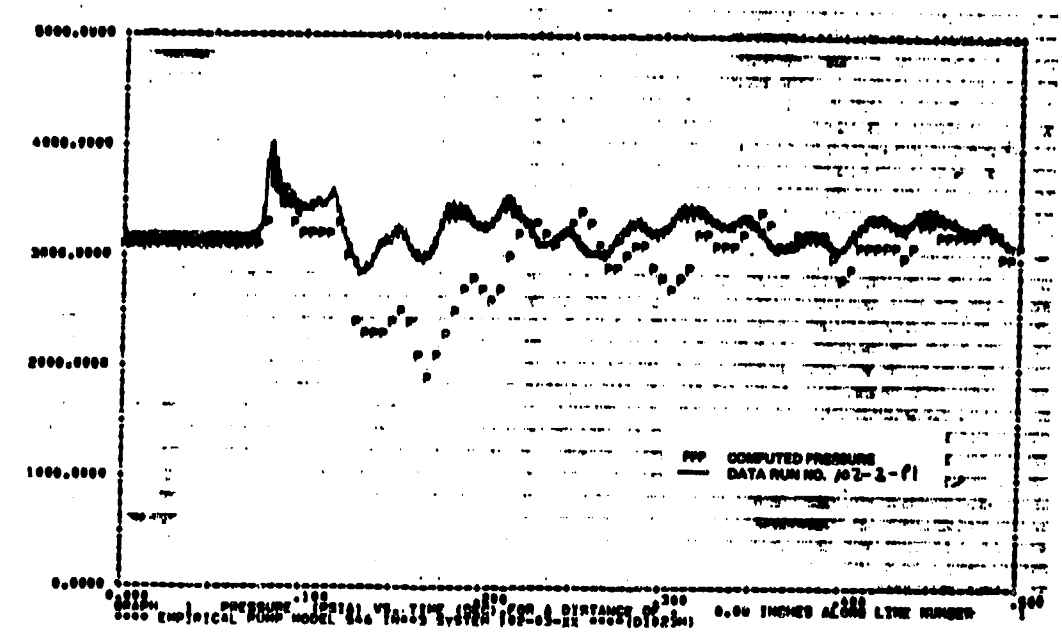


FIGURE 99 FIRST ORDER EMPIRICAL PUMP MODEL 102-3-P1 TRANSIENT SIMULATION

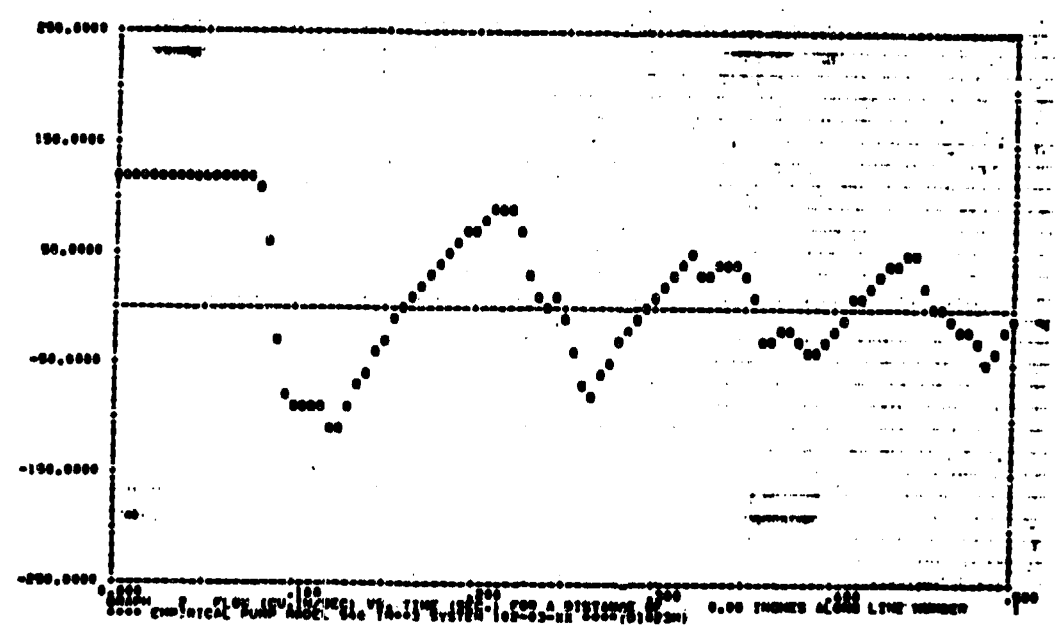


FIGURE 100 FIRST ORDER EMPIRICAL PUMP MODEL 102-3 PUMP OUTLET FLOW TURN-OFF TRANSIENT SIMULATION

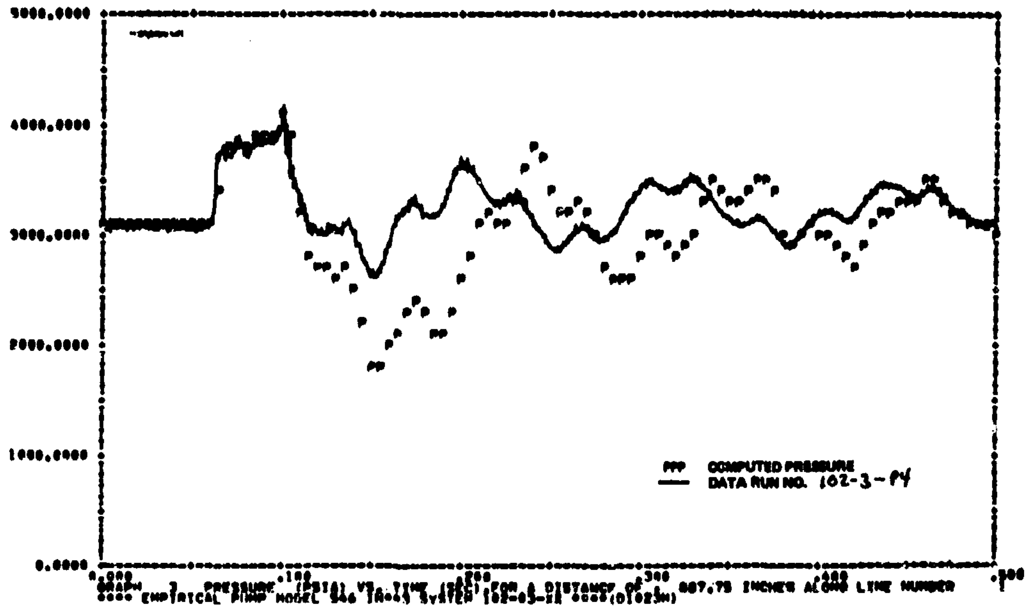


FIGURE 101 FIRST ORDER EMPIRICAL PUMP MODEL 102-3-P4 TRANSIENT SIMULATION

TABLE 13 SECOND ORDER EMPIRICAL PUMP MODEL INPUT DATA FOR TEST DATA RUN 103-04 TURN-OFF TRANSIENT

```
**** EMPIRICAL PUMP MODEL 37 IN**3 SYSTEM 103-04-XX ****(DRPUM57)
   .0005       .5        .005      100.
  2   3    1
  1                2          360          58.5      1.      .058   .3E+08
  2                                         40.      1.      .058   .3E+08
  1  57    2   -1    1    0
  1950.     4600.     225.      3165.      3100.    150.0     5.0      .04
    2.0      .004      10.
  2  23    3    1   -2                                        4
   .022      .65
     0.      .070      .072        .5
  .7756     .7756      .017      .017
  3  61    1    2
   100.
  3   2
  1   1    2    3      10.
  1   1    0    1    2    1
  2   2    3    2      10.
  0   2    3    1
  1   7    1   00                       2
  1   4        .01     -.01      58.0   -58.0
  1   1    1    2    1    3    1    4    1    5    1    6    1   14
  1        0.     5000.   3        0.   5000.
```

The period of torque oscillation is estimated from Figure 42. There are approximately 9 oscillations in .38 seconds or an average period of .0419 sec or 23.80 Hz. The natural damped circular frequency is 149 rad/sec. The settling time is the time required for the response curve to reach and stay within a range of the final value. The range can be 2% or 5% of the final value. The settling time is related to the largest time constant of the system and is a function of the system natural frequency (ω_n) and damping factor (ζ). For a given ω_n and a range of ζ between 0 and 1, the settling time for a lightly damped system is large compared to that for a properly damped system.

For an underdamped second order system the transient response to a unit step is bounded by a set of curves defined by equation (25).

$$1 \pm \left(e^{-\zeta \omega_n t} / \sqrt{1-\zeta^2} \right) \tag{25}$$

The time constant of these envelope curves is $T = \frac{1}{\omega_n \zeta}$. If the 5% tolerance band is applied for the settling time, then by applying equation (25), t_s is approximately three times the time constant or

$$t_s = 3T = \frac{3}{\omega_n \zeta} \tag{26}$$

The settling time from the drive torque response in Figure 100 is at least .5 sec. Also the damped frequency is related to the undamped natural frequency by equation (27).

$$\omega_d = \omega_n \sqrt{1-\zeta^2} \tag{27}$$

Substituting the known values into equations (26) and (27) results in two equations with unknowns ω_n and ζ. Making the appropriate substitutions and solving

$$\omega_n \zeta = \frac{3}{.5}$$

$$149. = \omega_n \sqrt{1-\zeta^2}$$

$$\zeta = .040$$

$$\omega_n = 150.\ \text{RAD/SEC}$$

The flow response delay is the difference between the start of the pump outlet pressure rise and the first change in drive torque. For the turn-off transient run the value was .004 seconds.

The input data file is shown in Table 13. The computed pump outlet pressures and flow are in Figures 102 and 103. The pressure data in Figure 102 matches the computed data in peak transient prediction. The resulting pressure oscillations between the measured and computed data have the same frequency. The test data changes frequency after the first transient spike and is also more damped. Another computer run was made increasing the damping factor ζ to .1. The predicted pressure in Figure 104 has the pressure magnitudes more in line with the overplotted data, even though the phasing is slightly different.

A turn-on transient was simulated with the second order flow model in the 37 in^3 system. The HYTRAN input data for the simulation is shown in Table 14. The drive torque in Figure 43 reflects a critical or overdamped response with a $\zeta \geq 1$. The second order function is invalid in this region. Therefore the damping factor was set to 0.9 in the simulation. The natural frequency is also undefined for damping ratios greater than or equal to one, but was computed using the established criteria. The settling time is .12 sec but the natural frequency cannot be found using Equation (27). Using Equation (26) the settling time reaches a minimum value around $\zeta = 0.68$ and increases linearly for larger ζ values. The discontinuity arises because an infinitesimal change in the value of ζ can cause a finite change in the settling time. Using the 5% band criteria the settling time for an almost critically damped system (ζ=.9) will be five times the time constant or

$$t_s \approx 5T = \frac{5}{\omega_n \zeta} \tag{28}$$

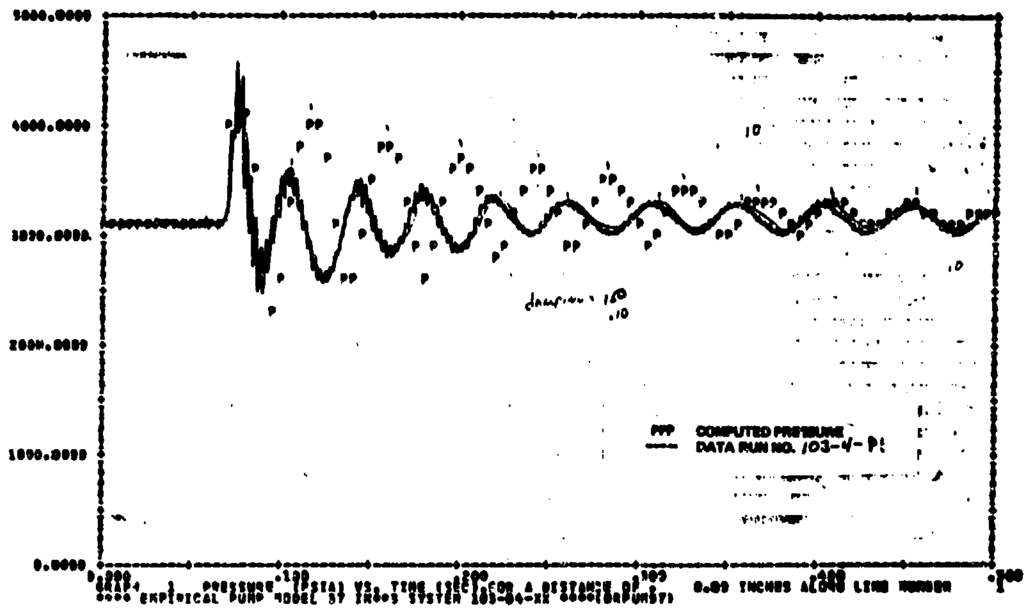


FIGURE 102 SECOND ORDER EMPIRICAL PUMP MODEL 103-4-P1
TRANSIENT SIMULATION WITH .04 DAMPING FACTOR

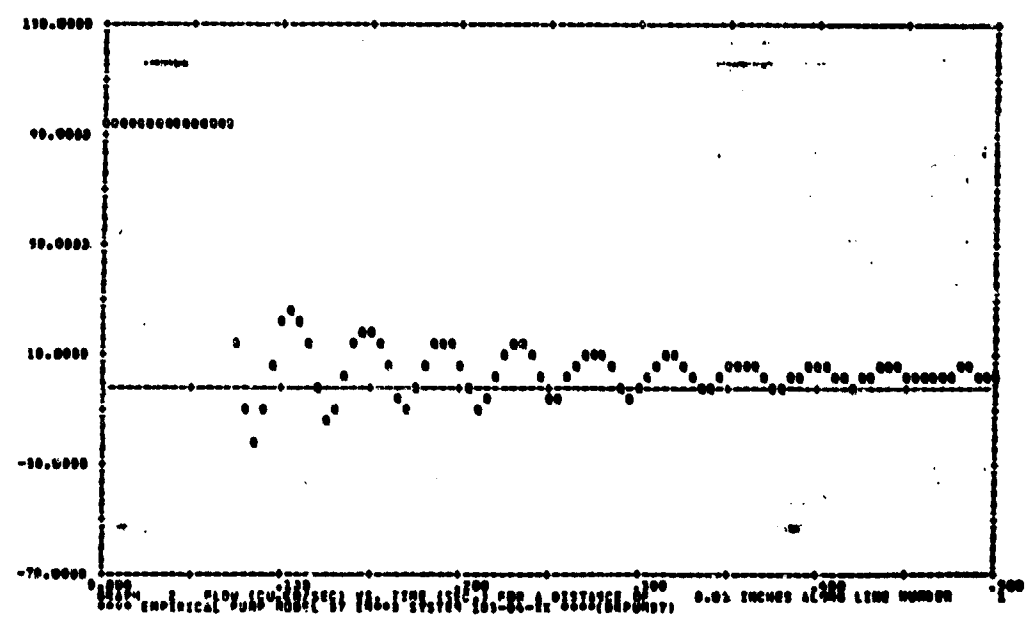


FIGURE 103 SECOND ORDER EMPIRICAL PUMP MODEL 103-4 PUMP
OUTLET FLOW TURN-OFF TRANSIENT SIMULATION

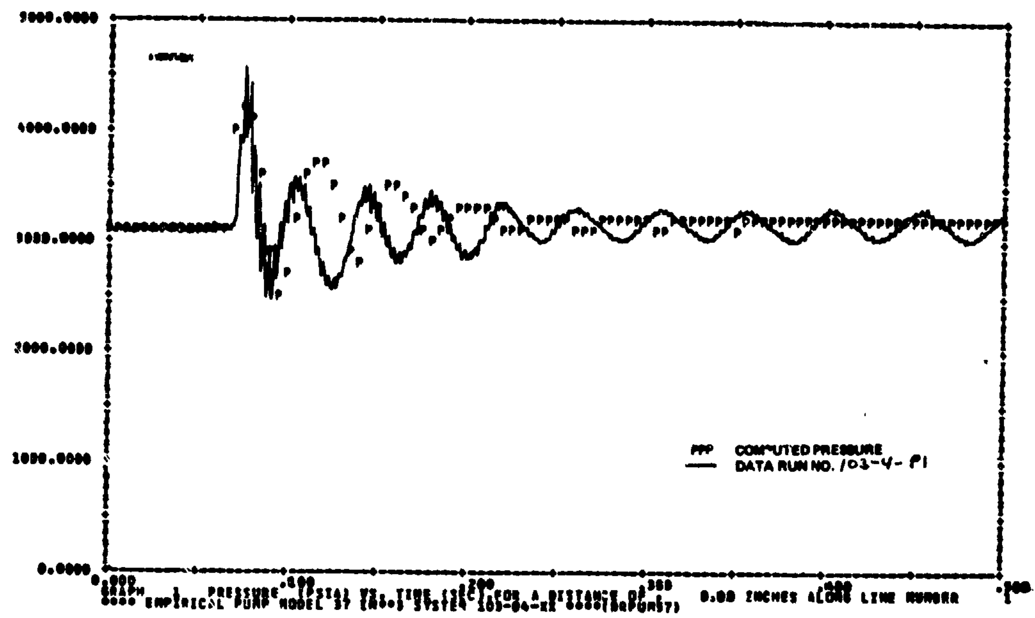


FIGURE 104 SECOND ORDER EMPIRICAL PUMP MODEL 103-4-P1 TRANSIENT SIMULATION WITH .1 DAMPING FACTOR

TABLE 14 SECOND ORDER EMPIRICAL PUMP MODEL INPUT DATA FOR TEST DATA RUN 103-4 TURN-OFF TRANSIENT

```
**** EMPIRICAL'PUMP MODEL 37 IN**3 SYSTEM 103-04+XX ****(DRPM57A)
   .0005        .5       .005       100.
  2    3    1
  1                 2        360                58.5         1.        .058      .3E+08
  2                                             100.         1.        .058      .3E+08
  1   57    2   -1    1    1
   1950.     4600.       225.      3165.       3100.       46.0         5.0       .9000
    95.0      .005        10.
  2   23    3    1   -2                                                  6
    .022       .65
      0.      .050       .080       .082        .105         .5
    .017      .017      .7756      2.000       .7756      .7756
  3   61    1    2
    100.
  3    2
  1    1    2    3        10.
  1    1    0    1    2    1
  2    2    3    2        10.
  0    2    3    1
  1    7    1   00                             02
  1    4       .01       -.01       58.0       -58 0
  1    1    1    2    1    3    1    4    1    5    1    6    1   14
  1         0.      5000.   3        0.       5000.
```

Applying equation (28) with the settling time of .12 seconds the undamped natural frequency is 46.0 RAD/SEC. The simulation results are overplotted with test data in Figures 105 and 106.

A turn-off transient computer run was made in a 280 in^3 system with the F-15 pump. The HYTRAN data file in Table 15 was used. The natural frequency was 82.45 rad/sec with a damping factor of .072. Figure 81 was used to obtain the settling time and undamped natural frequency. The pump outlet flow and pressures are shown in Figure 107 and 108. The pressure at the control valve 452" from the pump is shown in Figure 109.

A second order pump model was used to simulate a turn-on transient in a 538 in^3 system using the F-4 pump. The HYTRAN input data in Table 16 generated the computed pressures and flows in Figures 110, 111 and 112. The damping ratio was set to .9 and the settling time t_s was .5 seconds. The computed pressures match well with the test data.

A turn-off transient in the 538 in^3 system had a natural frequency of 37 RAD/SEC and a damping ratio of .16. The damped natural frequency was measured from Figure 62 as 36.5 RAD/SEC and the settling time was .51 seconds. The computed pump outlet pressure and flow are in Figures 113, 114 and 115.

d. Summary of Empirical Pump Input Data Requirements

Either HYTRAN empirical pump model can be used to simulate a particular test. The same input data will not work in a different system unless the programmer is aware of how to vary the input parameters. Multiple test systems should provide a better data base. The designer can extrapolate the information he measures to other system designs and perhaps to other pumps of similar capacity and response that he is unable to test.

The undamped natural frequency of the pump torque and hanger response are important parameters for both empirical pump models.

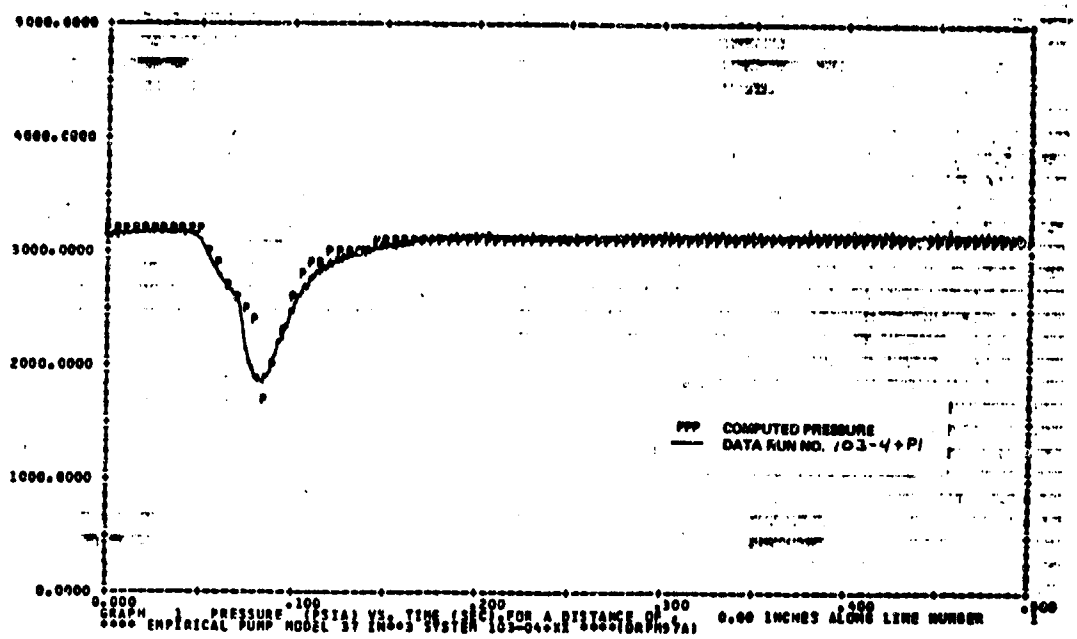


FIGURE 105 SECOND ORDER EMPIRICAL PUMP MODEL 103-4+P1 TRANSIENT SIMULATION

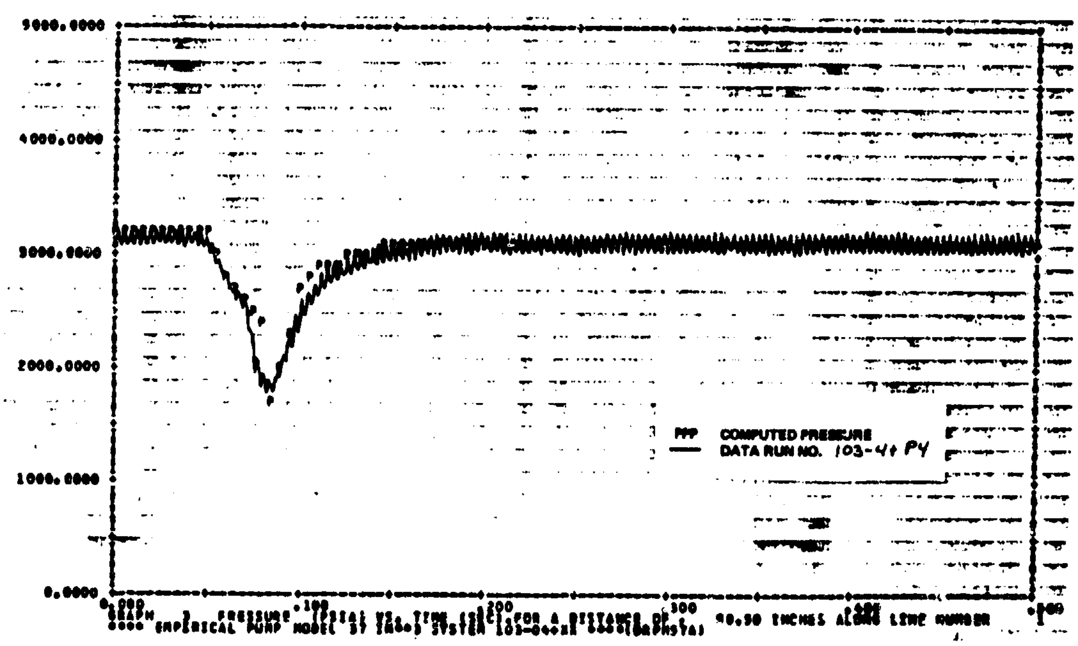


FIGURE 106 SECOND ORDER EMPIRICAL PUMP MODEL 103-4+P1 TRANSIENT SIMULATION

TABLE 15 SECOND ORDER EMPIRICAL PUMP MODEL INPUT DATA FOR TEST DATA RUN 100-2 TURN-OFF TRANSIENT

```
**** EMPIRICAL PUMP MODEL 280 IN**3 SYSTEM 100-02-XX ****(D1002M)
   .0005        .5       .005       150.
  2    3    1
  1                 2        360             452.        1.       .058    .3E+08
  2                                           40.        1.       .058    .3E+08
  1   57    2   -1    1
   3750.     4600.       225.      3165.     3100.     82.45       8.8     .0720
     2.0      .001        10.
  2   23    3    1   -2                                            4
    .022       .65
      0.      .078       .083         .5
   .6244     .6244       .017       .017
  3   61    1    2
    100.
  3    2
  1    1    2    3        10.
  1    1    0    1    2    1
  2    2    3    2        10.
  0    2    3    1
  1    5    1   00                         2
  1    4       .10      -.10       449.     -449.
  1    1    1    2    1    4    1    5    '    6
  1         0.     5000.   3          0.     5000.
```

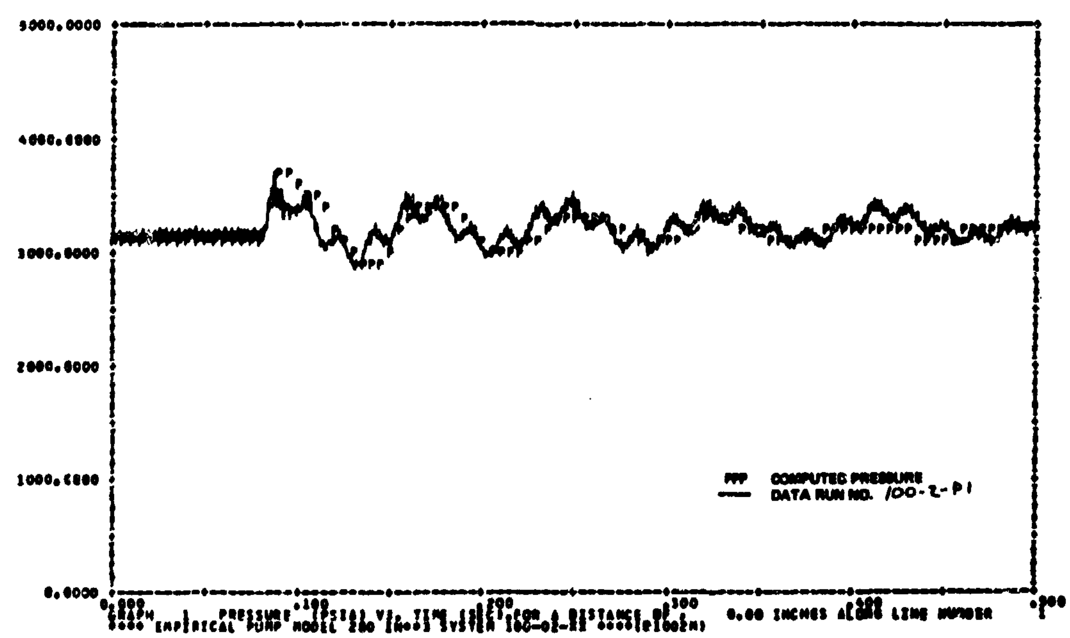


FIGURE 107 SECOND ORDER EMPIRICAL PUMP MODEL 100-2-P1 TRANSIENT SIMULATION

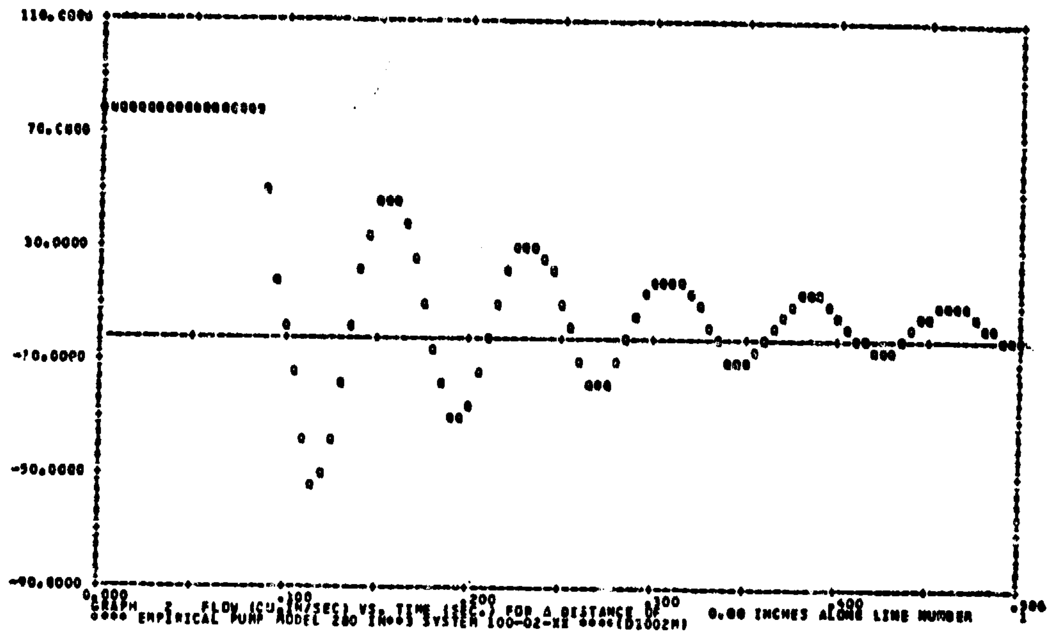


FIGURE 108 SECOND ORDER EMPIRICAL PUMP MODEL 100-2 PUMP OUTLET FLOW TURN-OFF TRANSIENT SIMULATION

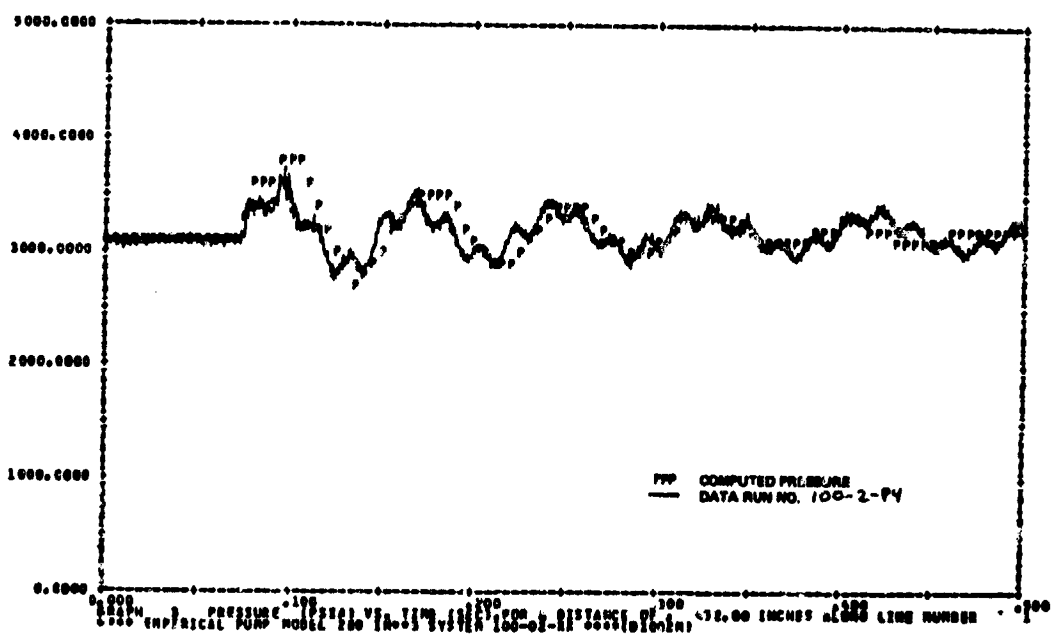


FIGURE 109 SECOND ORDER EMPIRICAL PUMP MODEL 100-2-P4 TRANSIENT SIMULATION

TABLE 16 SECOND ORDER EMPIRICAL PUMP MODEL INPUT DATA FOR TEST DATA RUN 105-04 TURN-ON TRANSIENT

```
**** EMPIRICAL PUMP MODEL 538 IN**3 SYSTEM 105-04+XX ****(D1054F)
    .001         1.         .01        148.
  2    3    1
  1                   2         360            876.75       1.        .058     .3E+08
  2                                              40.        1.        .058     .3E+08
  1   57    2   -1    1    1
   3750.     3750.      106.       3165.      3100.      18.51       3.8      .9000
    104.      .001       5.0
  2   23    3    1   -2                                                4
     022       .65
      0.      .050      .085         1.
    .017      .017     .8434      .8434
  3   61    1    2
    100.
  3    2
  1    1    2    3         10.
  1    1    0    1    2     1
  2    2    3    2         10.
  0    2    3    1
  1    5    1   00                          2
  1    4       .01       -.01       876.    -876.
  1    4    1    2    1    11    1    13    1    17
  1          0.      5000.    3          0.     5000.
```

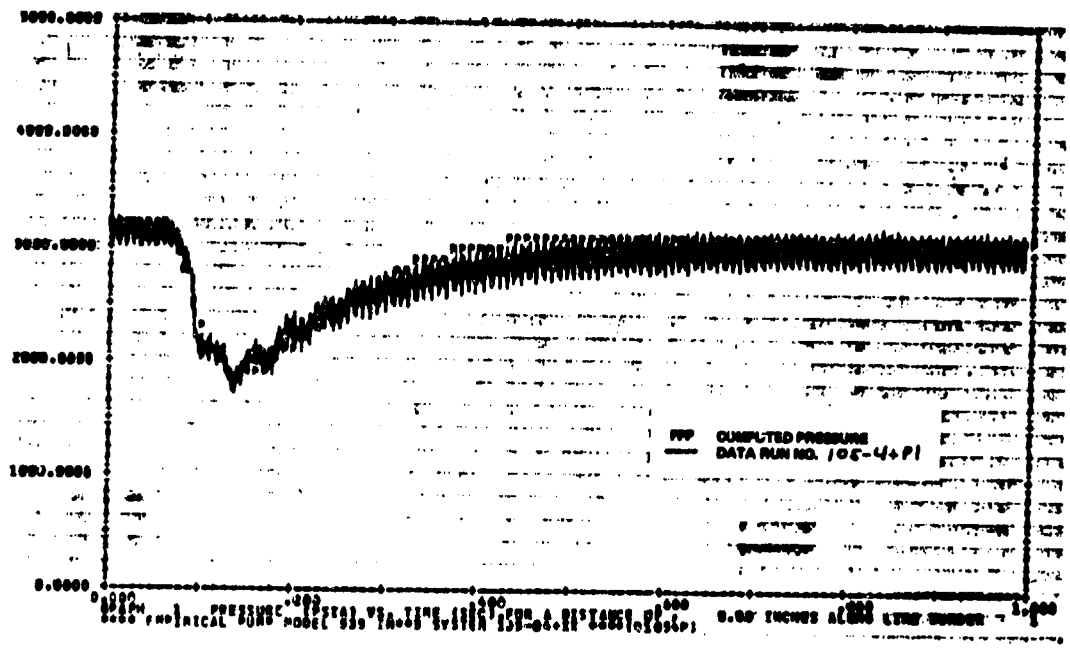


FIGURE 110 SECOND ORDER EMPIRICAL PUMP MODEL 105-4+P1 TRANSIENT SIMULATION

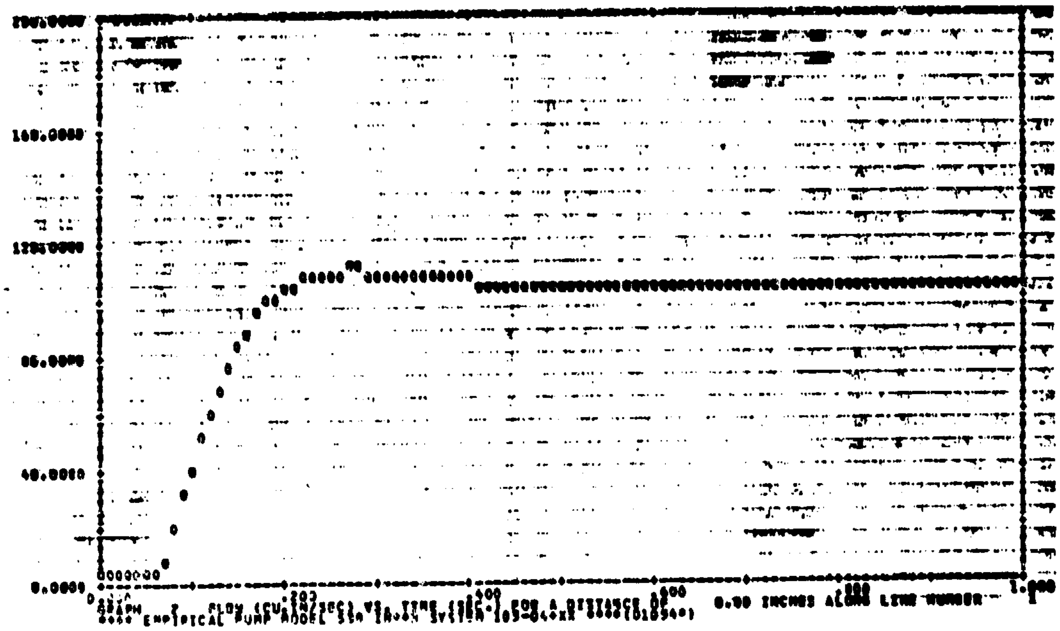


FIGURE 111 SECOND ORDER EMPIRICAL PUMP MODEL 105-4 PUMP OUTLET FLOW TURN-ON TRANSIENT SIMULATION

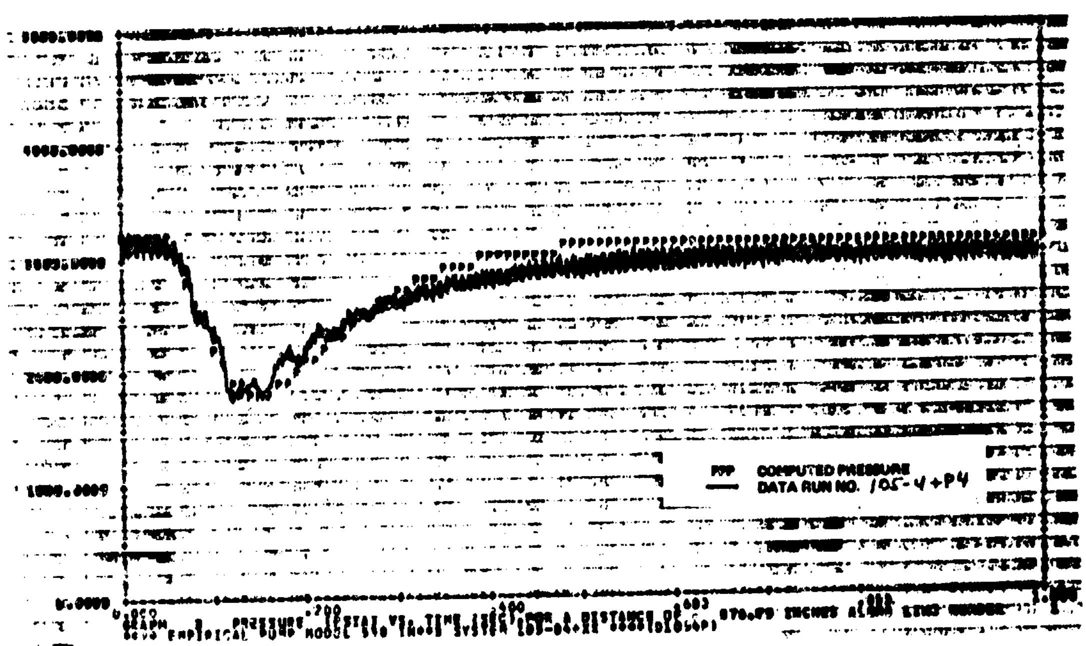


FIGURE 112 SECOND ORDER EMPIRICAL PUMP MODEL 105-4+P4 TRANSIENT SIMULATION

TABLE 17 SECOND ORDER EMPIRICAL PUMP MODEL INPUT DATA FOR TEST DATA RUN 105-04 TURN-OFF TRANSIENT

```
**** EMPIRICAL PUMP MODEL 538 IN**3 SYSTEM 105-04-XX ****(D1054M)
    0.001        1.        .01        145.
  2    3    1
  1                  2          360          876.75       1.        .058    .3E+08
  2                                           40.         1.        .058    .3E+08
  1   57    2   -1    1
   3750.      3750.       106.      3165.     3100.      37.0       3.8     .1600
     2.0       .001       5.0
  2   23    3    1   -2
    .022        .65
      0.       .070       .085        1.
   .8434      .8434       .017       .017
  3   61    1    2
    100.
  3    2
  1    1    2    3       10.
  1    1    0    1    2    1
  2    2    3    2       10.
  0    2    3    1
  1    5    1   00                           2
  1    4        .01      -.01       876.    -876.
  1    4    1    2    1   11    1   13    1   17
  1         0.       5000.   3         0.     5000.
```

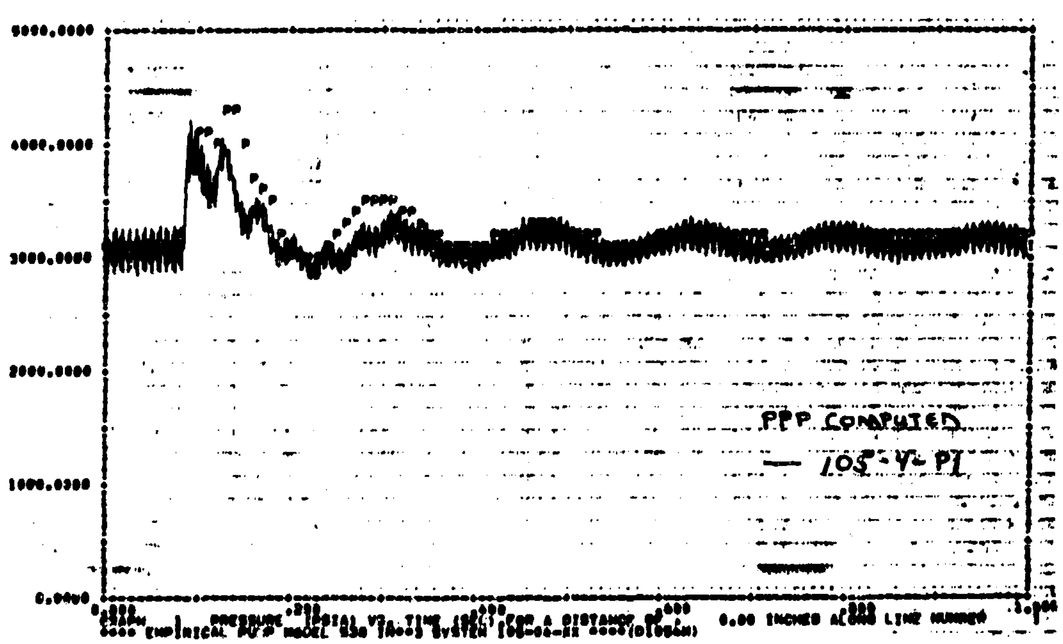


FIGURE 113 SECOND ORDER EMPIRICAL PUMP MODEL 105-4-P1 TRANSIENT SIMULATION

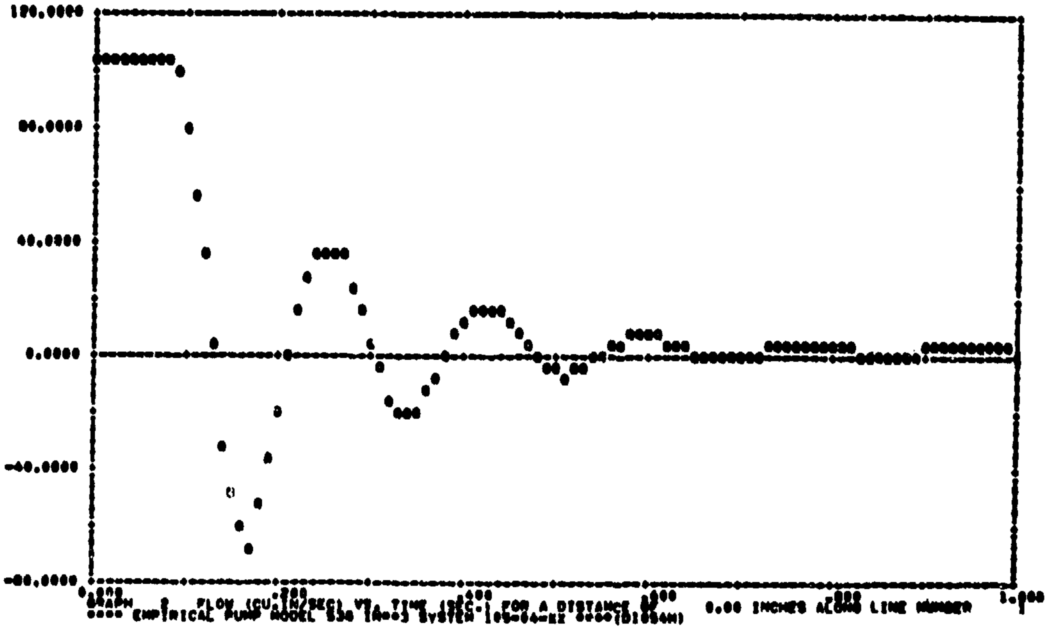


FIGURE 114 SECOND ORDER EMPIRICAL PUMP MODEL 105-4 PUMP OUTLET FLOW TURN-OFF TRANSIENT SIMULATION

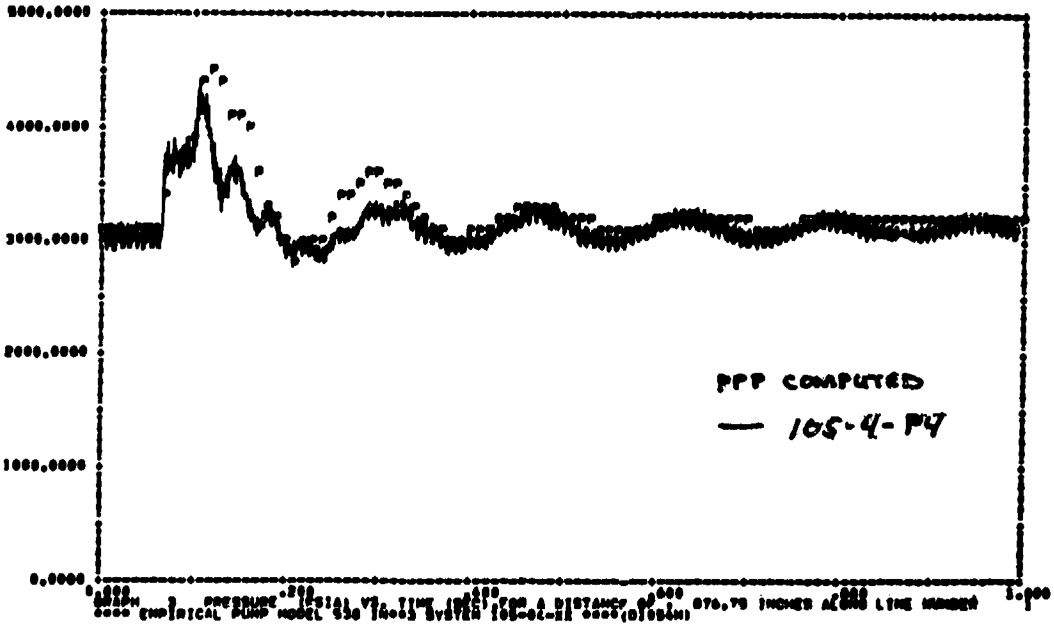


FIGURE 115 SECOND ORDER EMPIRICAL PUMP MODEL 105-4-P4 TRANSIENT SIMULATION

Plots of the ratio of actual flow over rated flow versus the natural frequency period τ in seconds of the drive torque are shown in Figures 116, 117 and 118 for the F-15 and F-4 hydraulic pumps in the three different test systems. Figure 117 shows that the damping period is relatively constant for each flow rate except the turn-on transient with the F-4 pump at rated flow. When the pump outlet pressure falls significantly below 3000 psi at rated conditions, the flow response is much slower due to compressibility effects in the pump/system.

The input data requirements are outlined in the following paragraphs for the first and second order pump models.

(1) First Order Pump Model

(a) Obtain steady state parameters from test or specification sheets

- Pump speed
- Pump rated speed
- Pump flow at rated speed
- Pressure at full flow
- Pressure at zero flow
- Case drain flow at rated flow and presssure

(b) Evaluate first order time constant by measuring the period of one torque oscillation then multiply by 0.632.

$$\tau_1 = \tau_{PERIOD} * 0.632 \qquad (29)$$

(c) To remove spurious pressure signals the outlet pressure can be phased through a lag network which effectively removes the oscillation frequency. The pressure time constant can be zero. Too large a constant can cause a mathematical instability.

NOTE: The mathematical subroutine used in the pump model provides an approximate difference form of a continuous transfer function.

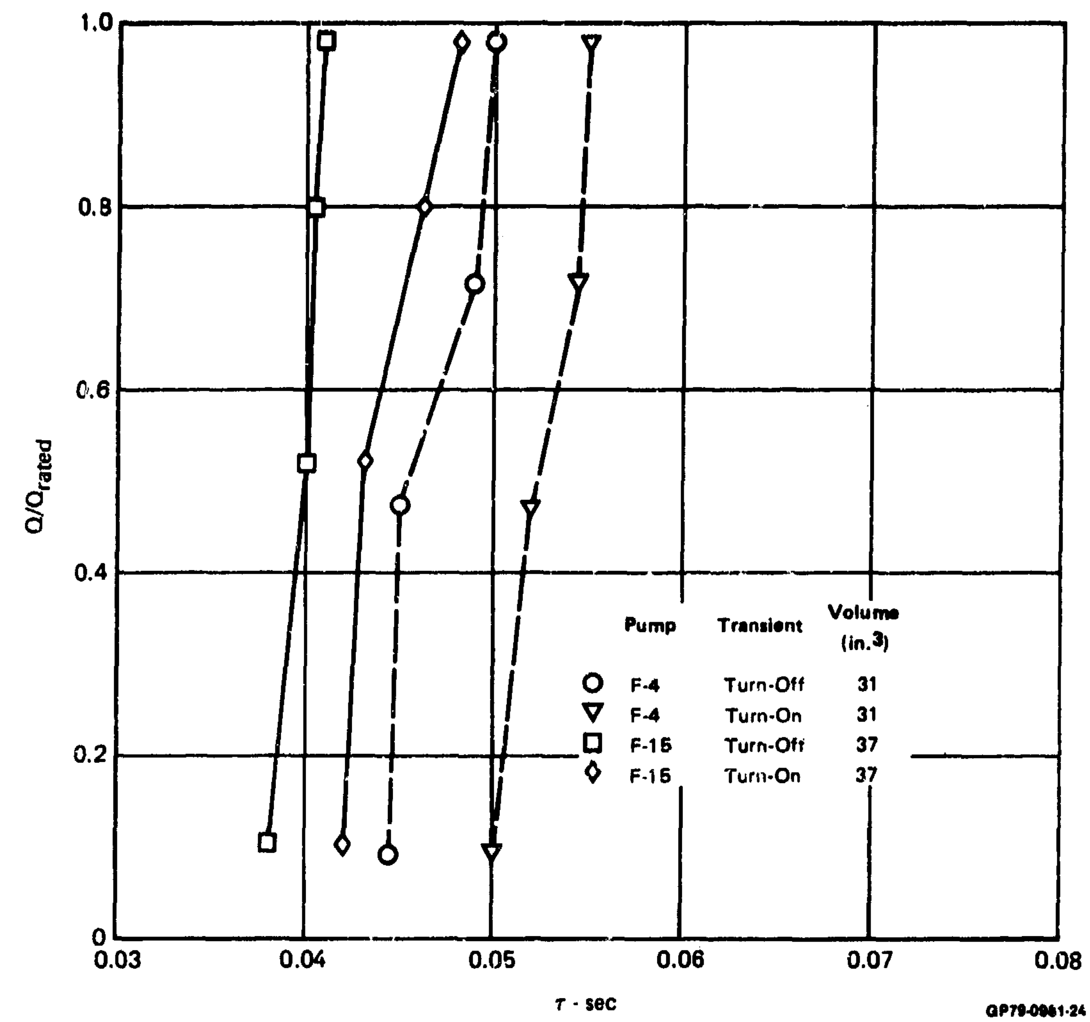


FIGURE 116 DAMPING PERIOD FOR 30 IN.^{3} SYSTEMS

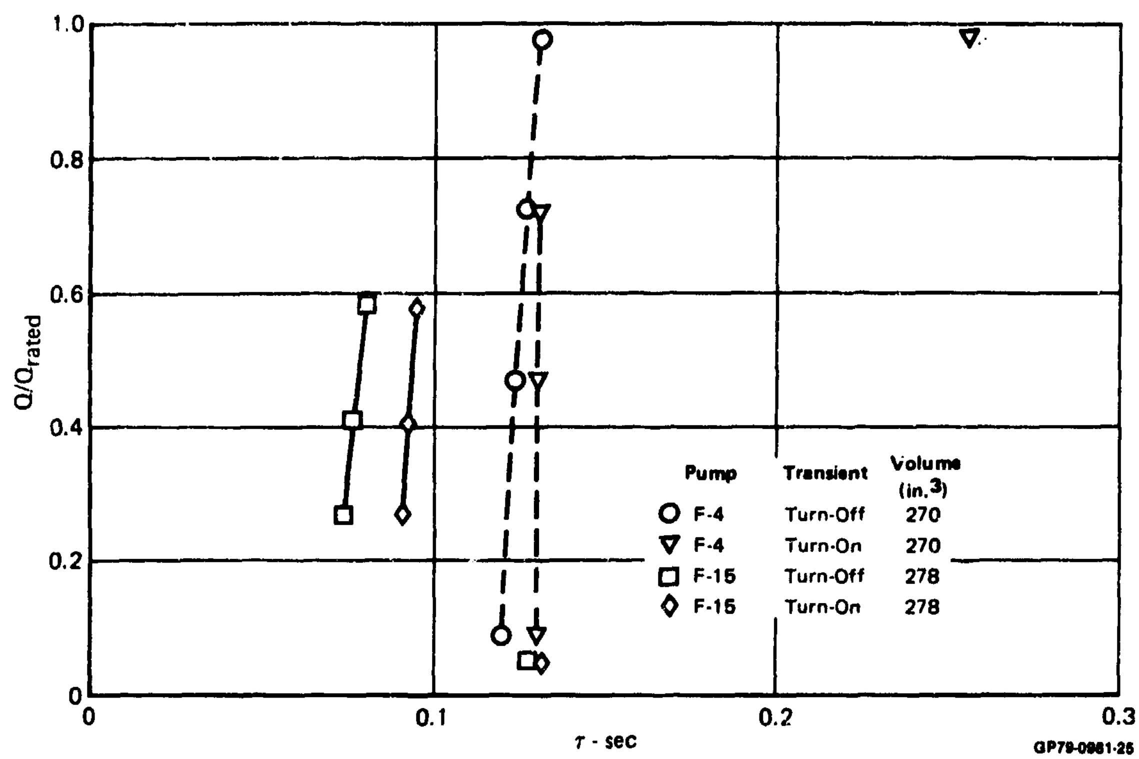


FIGURE 117 DAMPING PERIOD FOR 270 IN.^{3} SYSTEMS

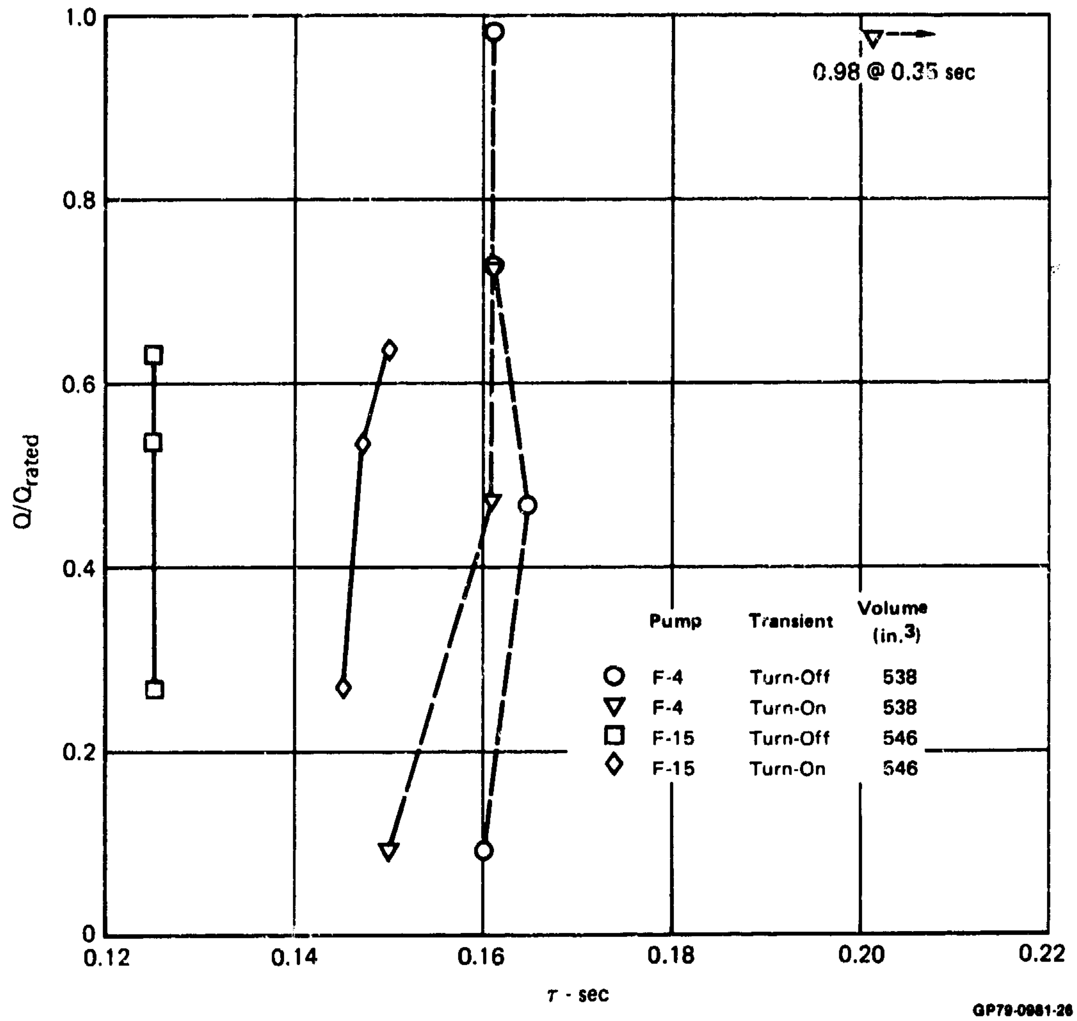


FIGURE 118 DAMPING PERIOD FOR 500 IN.^{3} SYSTEMS

The sample frequency (HYTRAN program calculation interval frequency) must be 10 to 15 times larger than the largest system frequency or discrepancies will arise.

(2) Second Order Pump Model

(a) Obtain steady state parameters from test or specification sheets

- Pump Speed
- Pump Rated Speed
- Pump Flow at Rated Speed
- Pressure at Full Flow
- Pressure at Zero Flow
- Case Drain Flow at Rated Flow and Pressure

(b) Evaluate the damped natural frequency period from the drive torque curve. Estimate the settling time as 5% of the final torque value. Use the following equations

$$t_s = \frac{3}{\omega_n \zeta} \tag{30}$$

$$\omega_d = \omega_n \sqrt{1 - \zeta^2} \tag{31}$$

To solve for the undamped natural frequency ω_n and the damping factor ζ.

(c) Input the desired pump outlet steady state flow rate after the transient.

(d) Measure the difference between the start of the pump outlet pressure transient and the start of the drive torque transient. Enter the difference as the delay time.

(e) Measure or estimate the pump outlet chamber volume.

Notes: Special programming of the second order pump model will be required to study reversal transients.

The delay time is not necessary, but an outlet volume must be provided.

The simulation can be improved with a measured valve closing time and accurate system volume.

3. HYTRAN BASIC PUMP MODEL

The HYTRAN basic pump model simulates the operation of a variable displacement, pressure compensated, hydraulic pump. Inlet, case, and outlet pressures and flows are calculated. A complex model requires many detailed input parameters and the HYTRAN model is no exception. However, many of the parameters require the user to be familiar with the pump operation and the relationships of key variables. The validity of a simulation depends on the user's ability verify the pump model input data. Oftentimes the data is difficult to obtain.

Unfortunately, the required input data cannot be reduced without compromising the computed output accuracy, but the user task can be greatly simplified. The contract work emphasis was to make the pump model more usable.

Some data for the current HYTRAN basic pump model had to be obtained from the manufacturer. The program input data was changed to allow the user to physically measure or scale from a drawing the required information. The dimensional data is then used in the pump subroutine to compute the input parameters. A detailed discussion is included to guide the user to derive the non-dimensional data. These changes should greatly enhance the usability of the pump model.

The HYTRAN pump model changes, input parameter derivation, and correlation with previously collected test data are discussed in the following paragraphs.

a. Program Changes

Modifications and enhancements were made to the HYTRAN basic pump model. The following sections describe the changes.

1. Orifice Coefficient

The fluid flowrate through an orifice neglecting the inlet velocity is

$$Q = Cd\ A \sqrt{\frac{2\Delta P}{\rho}} \qquad (32)$$

where

Q = orifice flow (CIS)

Cd = discharge coefficient

A = orifice area (in^2)

ΔP = pressure drop (PSID)

ρ = fluid density (lb-sec^2/in^4)

The inlet velocity can have an effect on the discharged flow through an orifice. The effects of inlet velocity are incorporated in Equation (33).

$$Q = \frac{Cd\ A}{\sqrt{1 - (\frac{do}{d1})^4}} \sqrt{\frac{2\ \Delta P}{\rho}} \tag{33}$$

where

do = orifice diameter (in)

d1 = inside line diameter upstream of the orifice (in)

The quantity $\frac{Cd}{\sqrt{1 - (\frac{do}{d1})^4}}$ is called the flow coefficient, C. The flow coefficient is a function of Reynolds number and the inlet to orifice diameter ratio $(\frac{do}{d1})$. The flow coefficient for square edged orifices is computed by the HYTRAN pump model for each compensator position. This eliminates the user having to input a discharge coefficient from pump outlet to actuator and actuator to case. Further discussion of the flow coefficient calculation can be found in Appendix D of AFAPL-TR-76-43 Volume VI.

2. Flow Force Calculation

Steady state flow forces on the compensator spool were rederived in the pump model. The net force acting on a spool land caused by the fluid accelerating through the vena contracta formed between the spool and sleeve can be broken into axial and lateral components.

The lateral component pushes the valve spool sideways against the sleeve and is usually compensated for by symmetrical location of valve ports around the spool. The axial force is not compensated and generally acts to close the valve port. Neglecting compressibility, a net force acting on the spool can be written as

$$F_{AXIAL} = 2. * Cd * W * \chi * \Delta P * \cos \Theta \qquad (34)$$

where

Cd = discharge coefficient

W = valve width (IN)

χ = slot opening (IN)

ΔP = pressure drop across orifice (PSI)

Θ = jet angle

(Equation (34) has been verified by Merritt, Blackburn et al). Equation (34) assumes that the orifice is rectangular and the peripheral width is large compared with its axial length. The jet angle for most valves is 69° and the discharge coefficient can be set to .6.

Equation (34) provides a jet steady state flow force which acts to close the compensator valve. In HYTRAN the flow force is computed based on the previous time step valve position and pressure drop. The force is translated to a displacement through the compensator spring rate and is subtracted from the latest compensator displacement.

Transient flow forces act as a stabilizing force when flowing from actuator to case because the forces tend to close the valve. Flow from outlet to the actuator tries to open the valve and is a destabilizing force. The transient flow force is due primarily to acceleration of fluid in the valve chamber. The magnitude of the force may be written as

$$F_T = Cd \; W \; \sqrt{2 \; \Delta P \rho} \; L \quad \frac{d\chi}{dt} \qquad (35)$$

The compensator valve velocity was typically 1 IN/SEC. The flow length for the destabilizing force was approximately 1.2 IN. Assuming Cd = .6 with a pressure drop of 2200 PSI at 120°F and the maximum slot opening, the transient flow force can be computed as

$$F_T = (.6)\ (.125)\ \sqrt{2\ (2200)\ (.814E-4)}\ \ (1.2)\ (1)$$

$$= \ .0538 \text{ LBS}$$

which is negligible. Obviously the actual transient flow forces are not simple to compute, but the effects are minor and the calculation was omitted from the HYTRAN pump model.

3. Offset Flow Calculation

The F-15 pump hanger is offset from the barrel centerline which contains the pumping elements. The net result of the offset is the generation of a small piston flow whenever the actuator is stroking the hanger. The old HYTRAN pump model required the user to input an offset coefficient. The model was revised to internally calculate the coefficient. The user is now required to enter the number of pistons (N), piston diameter offset distance (OFFSET) and the actuator lever arm (LEVACT). The OFFSET coefficient is then computed as

$$\text{COEOSO} = \frac{\text{OFFSET} * \text{N} * \text{A}_{\text{PISTON}}}{2 * \text{LEVACT}} \qquad (36)$$

4) Compensator Position Calculation

The F-15 hydraulic pump transient test data showed the compensator position was proportional to pump outlet pressure. The information was used to derive an equation to compute compensator position as linear function of outlet pressure just before the compensator valve opens from actuator to case. This establishes a reference point on the compensator spool. The user also inputs the total spool sleeve overlap.

The slope of the outlet pressure vs compensator position curve is the compensator spring rate referenced to the compensator area. The slope and the input data point are used to compute the compensator position at zero outlet pressure. The position and slope then define a linear equation which is used to compute valve displacement for any pump outlet pressure.

5) Derivation of Actuator Pressure

Once the compensator position has been determined the actuator pressure can be computed. Whenever, the compensator valve is open from actuator to case, the actuator pressures and flows are directly calculated. With the valve in the overlapped region, or open from outlet to actuator, the actuator pressure is computed via an iteration on the outlet pressure. The algorithm is explained in the HYTRAN pump model in Appendix D. A new portion was added to account for diametral leakage flows past the spool in the lapped region of compensator travel. Leakages from pressure to actuator and actuator to case through the compensator valve and actuator are computed.

6) Case Drain Flow Area

The case drain outlet on the F-15 pump is a quick disconnect type fitting. When the pump is engaged with the pump manifold, the fitting is opened. Three slots are [illegible] the flow path . The previous HYTRAN pump model acc[illegible] [illegible]r the slots and computed an orifice pressure/ flow coefficient Because most hydraulic pumps do not have this feature and the case drain outlet diameters can vary, a case drain outlet flow area has been added to the input data. The HYTRAN pump model will compute an orifice pressure flow coefficient based on the area, fluid density, and a discharge coefficient of 0.6.

b. Derivation of Input Data Parameters

Many of the HYTRAN basic pump model input data parameters can be obtained from physical measurements of a disassembled pump. The data requiring computation is explained in the following paragraphs.

1) Pressure at Which the Compensator Valve is Line to Line From Actuator to Case.

The HYTRAN pump model simulates compensator position as a direct function of pump outlet pressure. To establish the linear relationship the compensator position is required for an outlet pressure. Figure 119 establishes the coordinate system to be used.

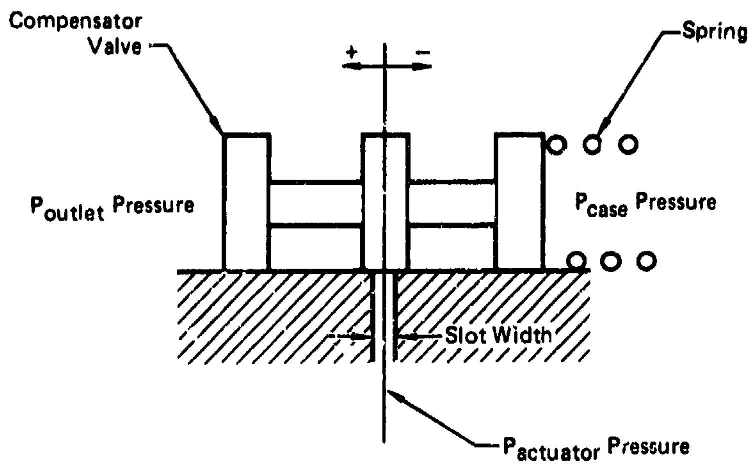


FIGURE 119 COMPENSATOR VALVE COORDINATE SYSTEM AND NOMENCLATURE

The following data is needed to compute the pressure:

Compensator Valve Area -------------------------	.15 IN^2
Spring Rate -------------------------------------	2000 LB/IN
Outlet Pressure @ a Known Flow (2 GPM) -------	2955 PSIA
Compensator Position @ 2 GPM with Respect to the Land Surface Exposed to the Case Pressure -	.0099 IN

The compensator position was taken from the position readout on the F-15 instrumented pump. The valve must move .0099 IN to be in a line to line position from actuator to case. Therefore the pressure can be computed from equation (37).

$$\text{POUTLET} - P_{\text{LINE TO LINE}} = \frac{\text{SPRING RATE}}{\text{AREA}} (.0099) \qquad (37)$$

$$P_{\text{LINE TO LINE}} = 2955 - \frac{2000}{.15} (.0099) = 2822 \text{ PSIA}$$

2) Compensator Slot Width

The pump model incorporates rectangular orifices. The slot width times the valve travel defines the flow area. Most valves have symmetrically located slots. The compensator slot width should be the width of one slot times the total number of slots.

3) Actuator Pressure Due to Spring Force at Zero Pump Displacement

The geometry of the problem is shown in Figure 120.

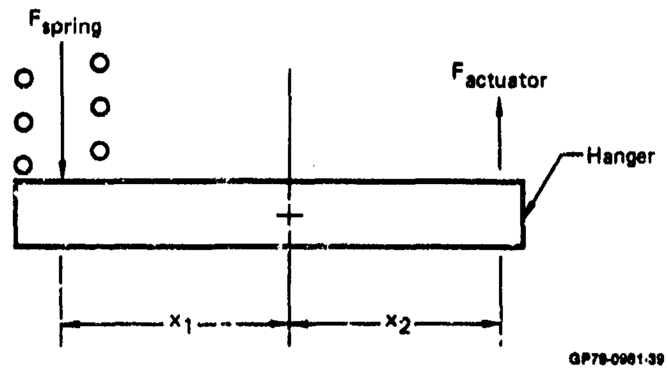


FIGURE 120 SPRING FORCE BALANCE ON HANGER

The following data is needed:

X_1 - - - - - - - - - - - - - - 2.7 in.
X_2 - - - - - - - - - - - - - - 2.07 in.

Spring Preload ------ 29 LB @ 2.1" Stroke
Spring Rate --------- 105 LB/IN
Spring Stroke at Zero Flow --- +1.26 IN
Actuator Area ------- .307 IN^2

The spring force is

$$F_S = 29.\ \text{lb} + \Delta F \qquad (38)$$

where

$$\Delta F = K\,(\Delta\chi) = 105\,(2.10 - 1.26) = 88.2\ \text{lb.}$$

Therefore

$$F_S = 29. + 88.2 = 117.2\ \text{lb.}$$

The actuator force can be obtained from the forces on the hanger and the respective lever arms.

$$F_S\,(2.7) = F_A\,(2.07)$$

$$F_A = 152.87\ \text{lb.}$$

or the actuator pressure is

$$P_{ACT} = \frac{152.87}{.307} = 497.94\ \text{PSI}$$

4) Actuator Pressure Due to Spring Force at Maximum Pump Displacement

For this calculation the lever arm lengths in Figure 120 change to

$$X_1 = 2.11 \text{ in.}$$
$$X_2 = 2.47 \text{ in.}$$

The force on the actuator is

$$F_A = \frac{29 * 2.11}{2.47} = 24.77 \text{ lb.}$$

or the pressure is

$$P_{ACT} = \frac{24.77}{.307} = 80.68 \text{ PSI}$$

5) Actuator Pressure Due to Piston Acceleration at 3600 RPM

Figure 121 shows the hanger geometry needed to compute the average piston acceleration.

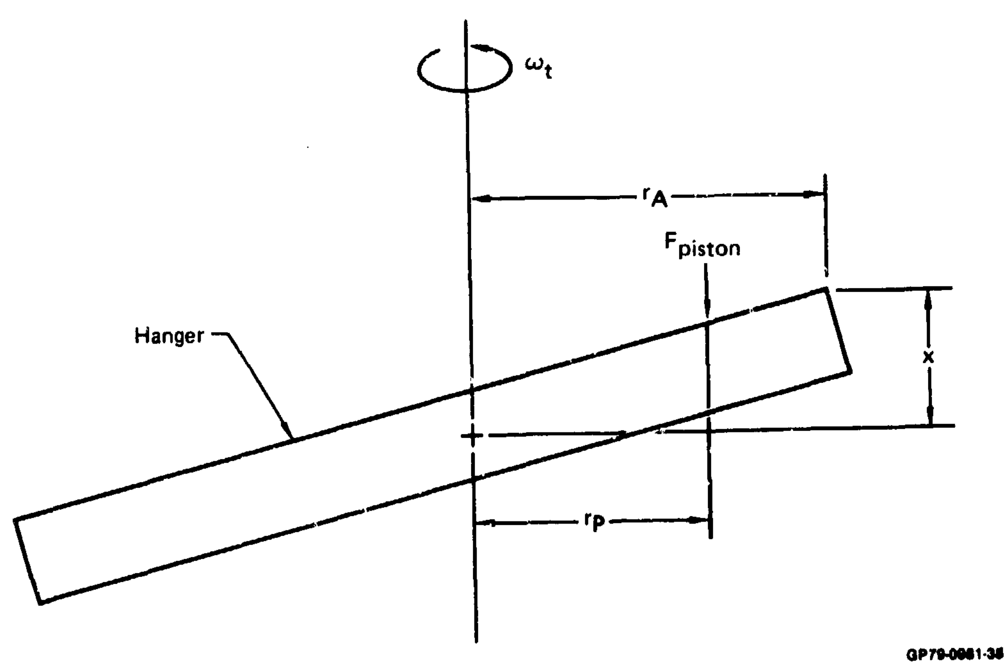


FIGURE 121 PISTON ACCELERATION FORCES ON HANGER

The following data is needed

rP = radial distance to piston centerline at max actuator displacement (1.172 IN)

rA = radial distance to actuator at max actuator displacement (2.47 IN)

χ = max actuator displacement (.795 IN)

w = piston rotational velocity $(3600 \frac{REV}{MIN} * \frac{2\pi}{60} = 377 \frac{RAD}{SEC})$

m = piston mass $(.0149 \text{ lb} * \frac{1}{386.4} \text{ in}^2/\text{sec} = 3.856\text{E-}5 \frac{\text{lb-sec}^2}{\text{IN}})$

A piston position for any time during a revolution is

$$Xp = \chi \frac{rp}{r_A} \text{SIN}(\omega t) \tag{39}$$

The piston velocity is

$$VP = \frac{\chi\omega rP}{rA} \cos \omega t \tag{40}$$

and the acceleration is

$$aP = -\chi \omega^2 \frac{rP}{rA} \sin \omega t \tag{41}$$

The piston force is equal to the mass times acceleration or

$$F_P = - m \chi\omega^2 \frac{rP}{rA} \sin \omega t \tag{42}$$

The torque on each piston is

$$T_P = -F_P * r' \tag{43}$$

where

$$r' = rP \sin \omega t$$

substituting equation (42) into equation (43) gives

$$T_P = m \chi_\omega 2 \frac{rP2}{rA} \sin^2 \omega t \tag{44}$$

The average torque per piston can be written as

$$T_{PAVE} = \frac{1}{2\pi} \int_o^{2\pi} T_P \, d\Theta \tag{45}$$

where $\Theta = \omega t$

Substituting equation (44) into equation (45)

$$T_{PAVE} = \frac{1}{2\pi} \int_0^{2\pi} (m\chi\, \omega^2 \frac{rP^2}{rA}) \sin^2 \Theta \, d\Theta \qquad (46)$$

But $\sin^2 \Theta = \frac{1-\cos 2\Theta}{2}$

So
$$T_{PAVE} = \frac{1}{2\pi} \int_0^{2\pi} m\chi\, \omega^2 \frac{rP^2}{r_A} \frac{1}{2} d\Theta - \frac{1}{2\pi} \int_0^{2\pi} m\chi\, \omega^2 \frac{rP}{rA} \frac{1}{2} \cos 2\Theta \, d\Theta$$

and evaluating the integral

$$T_{PAVE} = \frac{m\chi\omega^2 rP^2}{2r_A} \qquad (47)$$

Since $F_A \; r_A = F_P \; r_P$

The average total force for nine pistons is then

$$F_{AVE} = \frac{9m\chi\omega^2 r_P^2}{2 \; r_A^2} \qquad (48)$$

Substituting the appropriate numbers

$$F_{AVE} = \frac{9\,(3.856E\text{-}4)\,(.795)\,(377)^2(1.172)^2}{2\,(2.47)^2} = 44.142 \text{ lb.}$$

Referenced to the actuator piston area the actuator pressure is

$$P_{ACC} = \frac{44.142}{.307} = 143.78 \text{ PSI}$$

6) Actuator Pressure at 3600 RPM and Zero Pump Displacement

Figure 122 shows average hanger torque at 3600 RPM due to piston pressures as a function of hanger angle, steady state outlet pressure, and oil temperature. Torque due to pumping piston inertial forces are also shown. These torques along with the hanger spring act in the direction of full stroke (maximum hanger angle). The values in Figure 122 were generated by the HSFR pump model using various flow demands, system steady state pressures and fluid temperatures as the controlled variables.

At zero pump displacement (zero hanger angle) the hanger torque is 365 IN-LB. The lever arm between the hanger centerline and actuator is 2.07 IN. Consequently the force at the actuator is

$$\frac{365 \text{ in-lb}}{2.07 \text{ in.}} = 176.32 \text{ lb.}$$

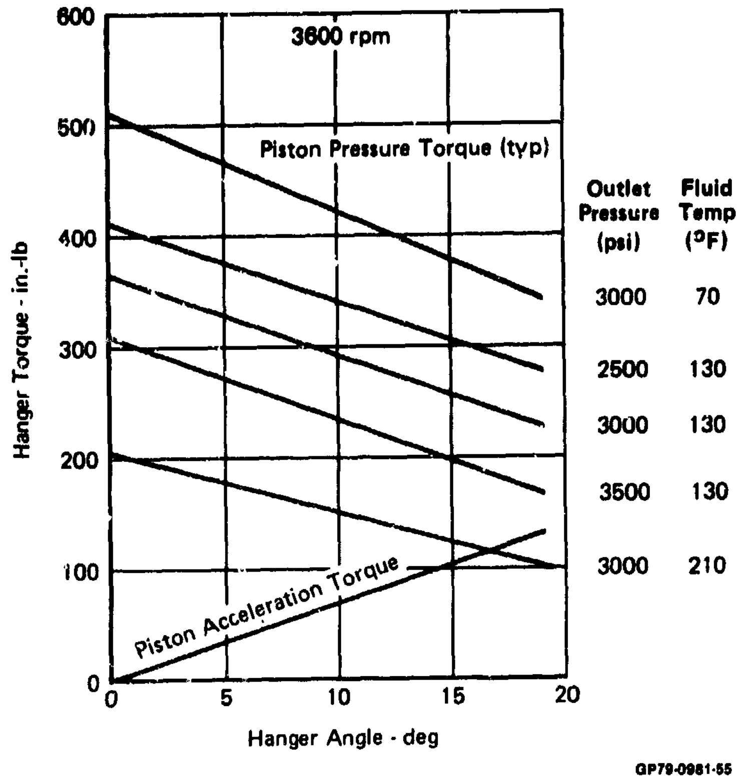


FIGURE 122 F-15 HYDRAULIC PUMP HANGER TORQUE

The actuator pressure at 3600 RPM is

$$P_{ACT} = \frac{176.32}{.307} = 574.33 \text{ PSI}$$

7) Actuator Pressure at 3600 RPM and Maximum Pump Displacement

From Figure 122 the torque at maximum pump displacement is 225 IN-LB. The lever arm is 2.47 IN and the force on the actuator is 91.09 LB. Reference to the actuator are, the actuator pressure is then

$$\frac{91.09}{.307} = 296.71 \quad \text{PSI}$$

8) Slope of Actuator Pressure Versus the Pump RPM Curve

The HSFR program was used to generate the actuator pressure versus RPM curve in Figure 123. The local slope at 3600 RPM is .035 PSI/RPM.

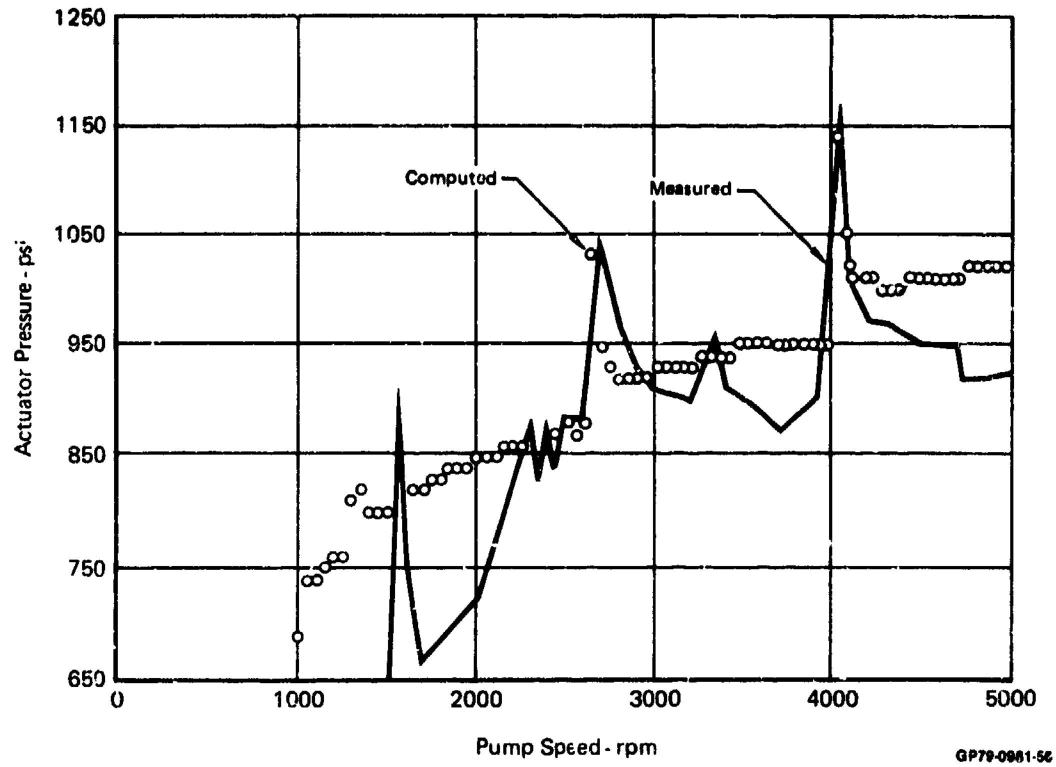


FIGURE 123 HANGER ACTUATOR CONTROL PRESSURE F-15 PUMP VERIFICATION TEST - SHORT LINE MIL-H-5606B, 3000 PSI, 110°F, 2.3 GPM

9) Hanger Damping

An estimate of hanger damping can be derived by assuming the pump to be a critically damped mass, spring, dashpot system. The damping coefficient (B) can be expressed by equation (49) in terms of the compensator spring rate and fluid and hanger mass.

$$B = 2\sqrt{KM} \tag{49}$$

where

B = damping coefficient (LB-SEC/IN)

K = spring rate (2000 LB/IN)

M = mass (.029 LB-SEC^2/IN)

Substituting the appropriate values into equation (49) the hanger damping is 15.23 $\frac{\text{LB-SEC}}{\text{IN}}$ or 50.0 PSI/IN/SEC when referenced to the actuator area of .307 IN^2.

The hanger damping term can vary depending on the system size or type of transient.

C. Correlation with Test Data

Data from turn-off and turn-on transients were compared to the HYTRAN simulation using the improved pump model. The HYTRAN input data file for a turn-off transient is shown in Table 18. The initial steady state flow rate was 77 CIS and the system temperature was 130°F. The results of the simulation are overplotted with test data in Figures 124 through 129.

The HYTRAN input data for a turn-on transient is presented in Table 19. The overplotted computer graphs are shown in Figures 130 through 135.

TABLE 18 HYTRAN INPUT DATA FOR RUN 94-A4-XX

```
****DATA RUN NO. 94-A4-XX  F-15 PUMP****(DPTST27)

THE TRANSIENT RESPONSE IS FROM T=0.0  TO T=   .200 SECONDS AT TIME INTERVALS OF DELT= .00020
      WITH OUTPUT POINTS PLOTTED AT INTERVALS OF   ,   .00200  SECONDS

FLUID DATA FOR  MIL-H-5606  AT  3000.0 PSIG, -    50.0 PSIG AND 130.0 DEG F  IN   10.0 DEG F STEPS
                    VISCOSITY    -  .100E-01       .140E-01IN**2/SEC
                    DENSITY      -  .813E-04       .803E-04(LB-SEC**2)/IN**4
                    BULK MODULUS -  .223E+06       .187E+06PSI
                    VAPOUR PRESS.-  .200E+01 AT  130.0 DEG F
FIX-UP TAKEN AT LINE 10,VEL OF SOUND IN LINE   6  IS    59.6PER CENT IN ERROR
FIX-UP TAKEN AT LINE 10,VEL OF SOUND IN LINE   7  IS    58.0PER CENT IN ERROR
FIX-UP TAKEN AT LINE 10,VEL OF SOUND IN LINE   8  IS    59.3PER CENT IN ERROR
```

LINE DATA LINE NO.	LENGTH	INTERNAL DIA	WALL THICKNESS	MODULUS OF ELASTICITY	DELX	CHARACTERISTIC IMPEDANCE	VELOCITY OF SOUND
1	53.7500	.9020	.0490	.300E+08	11.1500	6.2434	49088.3263
2	10.0000	.3190	.0200	.300E+08	10.0000	50.9780	50132.3294
3	302.7500	.8840	.0580	.300E+08	10.0724	6.5619	49551.4538
4	10.5000	.8840	.0583	.300E+08	10.5000	6.5619	49551.4538
5	10.9000	.4440	.0240	.300E+08	10.9000	25.9624	49460.3746
6	4.0000	.4440	.0200	.300E+08	4.0000	25.9624	20000.0000
7	4.1250	.9020	.0490	.300E+08	4.1250	6.2434	20625.0000
8	4.0000	.9020	.0490	.300E+08	4.0000	6.2434	20000.0000
9	192.0000	.9020	.0490	.300E+08	10.1053	6.2434	49088.3263

```
 COMPO.    1  INTEGER DATA       1  61   1  -1   0   0   0   0   0   0   0   0   0   0   0   0
REAL DATA CARD #    1    .6500E+02  0.        0.        0.        0.        0.        0.        0.
 COMPO.    2  INTEGER DATA       2  51   4   1  -3  -2   0   0   0   0   0   0   0   0   3   0
REAL DATA CARD #    1    .2022E+04  .2000E+04  .1500E+00  .2500E+00  0.        .1600E-01  .3826E+06  .2270E+01
REAL DATA CARD #    2    .3070E+00  .4980E+03  .8073E+02  .1430E+03  .5743E+03  .2967E+03  .3500E-01  .3900E+02
REAL DATA CARD #    3    .3000E+02  .7954E+00  -.3000E+00  .7000E-02  .2100E+01  .300E-02  .1097E+00  .4800E+02
REAL DATA CARD #    4    .5000E+31  .4603E+04  .6000E-01  .8000E-01  .4455E-01  .1040E-01  .1000E+01  .8000E+01
 COMPO.    3  INTEGER DATA       3  61   1   2   0   0   0   0   0   0   0   0   0   0   0   0
REAL DATA CARD #    1    .1000E+03  0.        0.        0.        0.        0.        0.        0.
 COMPO.    4  INTEGER DATA       4  11   3   3  -4  -5   0   0   0   0   0   0   0   0   0   0
 COMPO.    5  INTEGER DATA       5  21   5   4  -7   0   0   0   0   0   0   0   0   4   0   0
REAL DATA CARD #    1    .2200E-01  .6503E+30  0.        0.        0.        0.        0.        0.
REAL DATA CARD #    2    0.        .1003E-01  .2060E-01  .2000E+00  0.        0.        0.        0.
REAL DATA CARD #    3    .4005E+01  .4005E+01  0.        0.        0.        0.        0.        0.
 COMPO.    6  INTEGER DATA       6  41   1   5  -6   0   0   0   0   0   0   0   0   0   0   0
REAL DATA CARD #    1    .2100E-01  .6503E+00  0.        0.        0.        0.        0.        0.
 COMPO.    7  INTEGER DATA       7  11   0   6   7  -8   0   0   0   0   0   0   0   0   0   0
 COMPO.    8  INTEGER DATA       8  41   1   8  -9   0   0   0   0   0   0   0   0   0   0   0
REAL DATA CARD #    1    .1300E+00  .6503E+00  0.        0.        0.        0.        0.        0.
 COMPO.    9  INTEGER DATA       9  61   1   9   0   0   0   0   0   0   0   0   0   0   0   0
REAL DATA CARD #    1    .6500E+02  0.        0.        0.        0.        0.        0.        0.
      CPU TIME IN SECONDS =      1.365
```

```
                         STEADY STATE INPUT DATA
     NUMBER OF NODES =  6     NUMBER OF LEGS =  6     NUMBER OF CONSTANT PRESSURE NODES =  0
                         LEG CONNECTION INPUT DATA
LEG NO     UPST NODE NO    DNST NODE NO    NO OF ELEMENTS    FLOW GUESS    UPST PRESS    DNST PRESS
[illegible]

LEG NO     ELEMENTS IN LEG
[illegible]
```

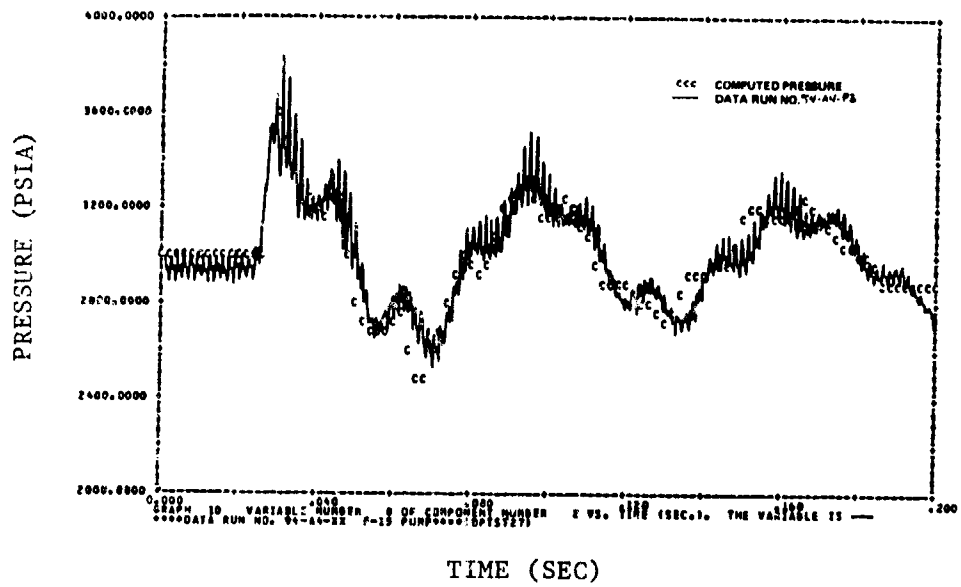


FIGURE 124 PUMP OUTLET PRESSURE RUN NO. 94-A4-P3

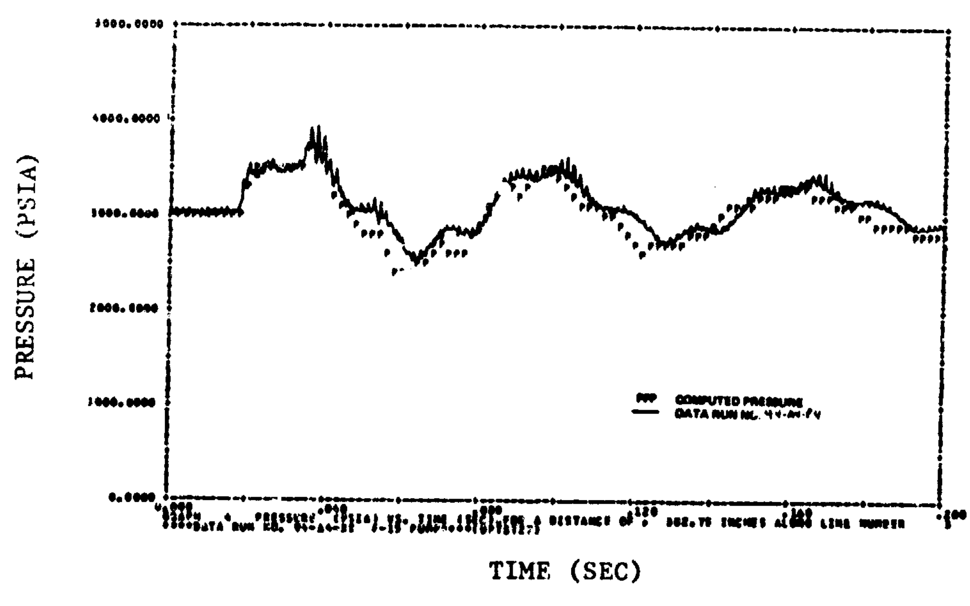


FIGURE 125 PUMP OUTLET PRESSURE RUN NO. 94-A4-P4

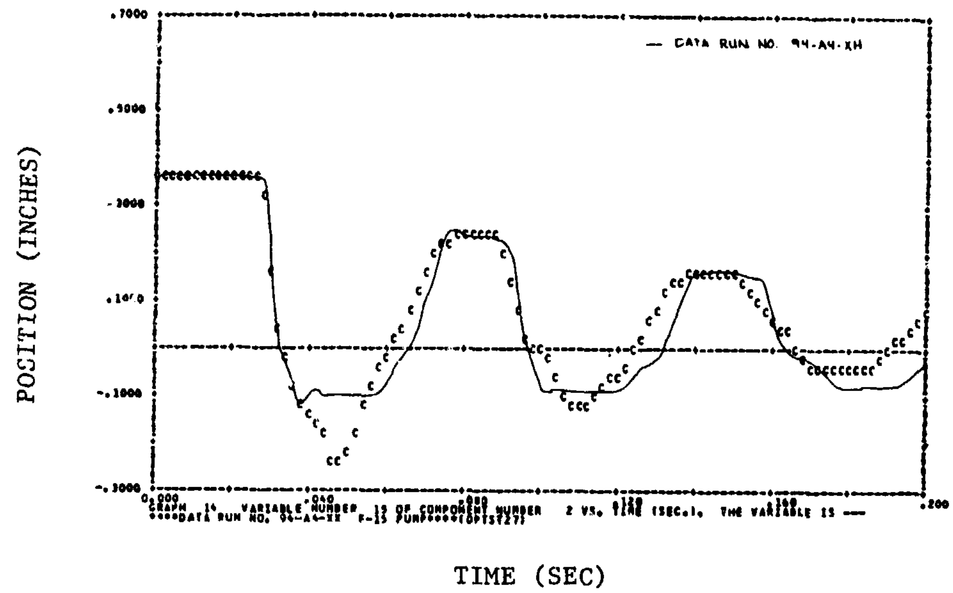


FIGURE 126 HANGER POSITION RUN NO. 94-A4-XH

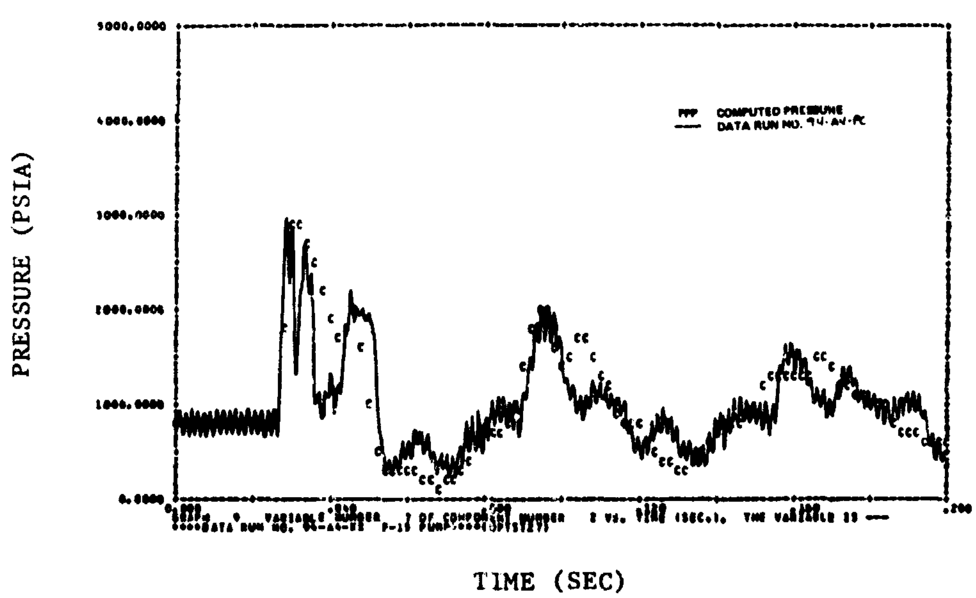


FIGURE 127 ACTUATOR PRESSURE RUN NO. 94-A4-PC

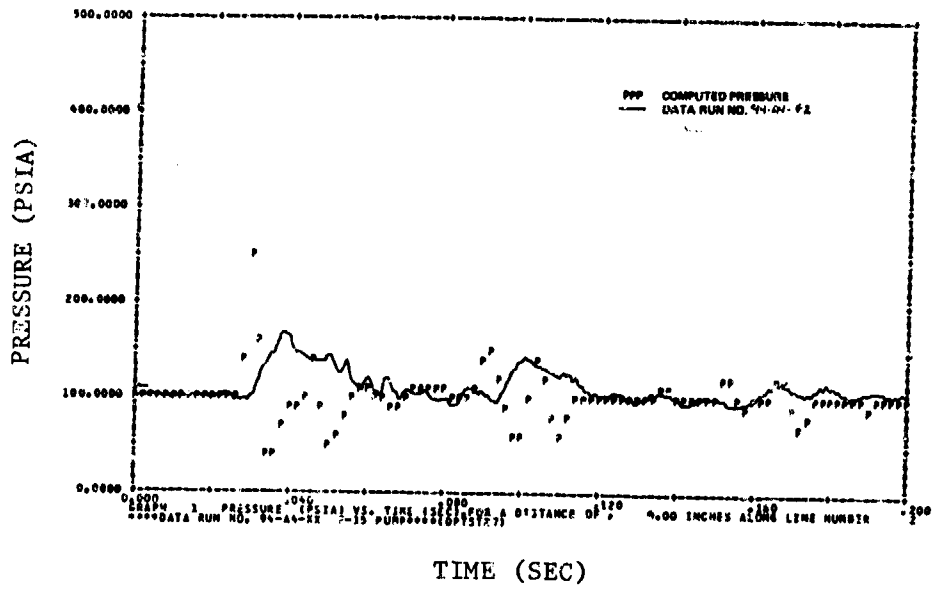


FIGURE 128 PUMP CASE PRESSURE RUN NO. 94-A4-P2

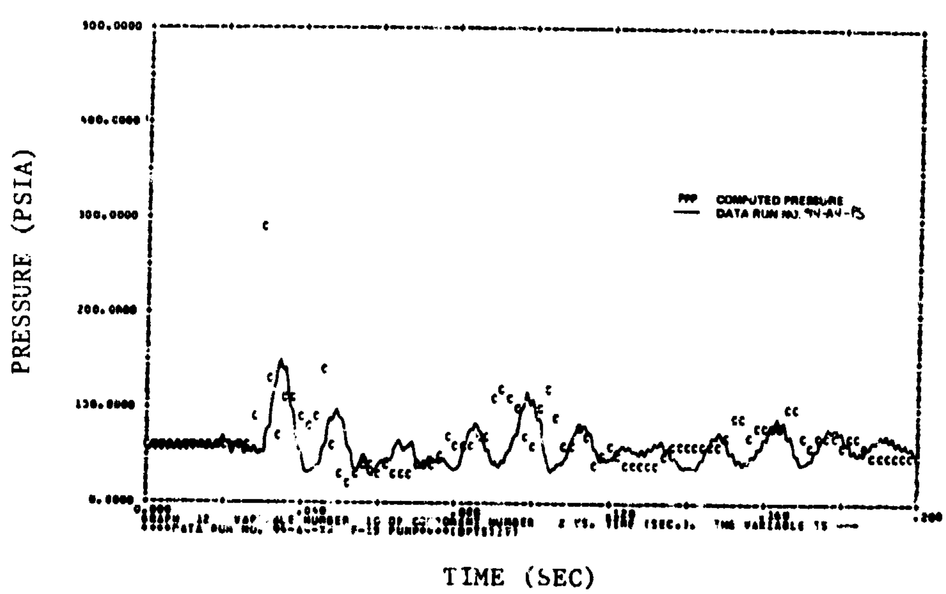


FIGURE 129 INLET PRESSURE RUN NO. 94-A4-P5

TABLE 19 HYTRAN INPUT DATA FOR RUN 94-A4+XX

```
****DATA RUN NO. 94-A4+XX  F-15 PUMP****(OPTSTZ8)

THE TRANSIENT RESPONSE IS FROM T=0.0  TO T=   .200 SECONDS AT TIME INTERVALS OF DELT= .00020
     WITH OUTPUT POINTS PLOTTED AT INTERVALS OF   ,   .00200  SECONDS

FLUID DATA FOR  MIL-H-5606  AT  3000.0 PSIG, -    50.0 PSIG AND 130.0 DEG F  IN   10.0 DEG F STEPS
                VISCOSITY     -  .186E-01        .146E-01IN**2/SEC
                DENSITY       -  .813E-04        .803E-04(LB-SEC**2)/IN**4
                BULK MODULUS  -  .223E+06        .187E+06PSI
                VAPOUR PRESS.-   .200E+01 AT  130.0 DEG F
FIX-UP TAKEN AT LINE 10,VEL OF SOUND IN LINE   6  IS    59.6PER CENT IN ERROR
FIX-UP TAKEN AT LINE 10,VEL OF SOUND IN LINE   7  IS    58.0PER CENT IN ERROR
FIX-UP TAKEN AT LINE 10,VEL OF SOUND IN LINE   8  IS    59.3PER CENT IN ERROR
```

LINE DATA LINE NO.	LENGTH	INTERNAL CSA	WALL THICKNESS	MODULUS OF ELASTICITY	DELX	CHARACTERISTIC IMPEDANCE	VELOCITY OF SOUND
1	55.7500	.9020	.0490	.300E+08	11.1500	6.243	49080.3263
2	18.0000	.3190	.0280	.300E+08	18.0000	50.9788	50132.3294
3	302.7500	.8840	.0580	.300E+08	10.0774	6.5615	49951.4530
4	10.5000	.8840	.0580	.300E+08	10.5000	6.5615	49951.4530
5	10.9000	.4440	.0280	.300E+08	10.9000	25.9624	49460.3746
6	4.0000	.4440	.0280	.300E+08	4.0000	25.9624	20000.0000
7	4.1250	.9020	.0490	.300E+08	4.1250	6.2434	20625.0000
8	4.0000	.9020	.0490	.300E+08	4.0000	6.2434	20000.0000
9	197.0000	.9020	.0490	.300E+08	10.1055	6.2434	49080.3263

```
 COMP#.   1  INTEGER DATA     1   61   1  -1   9   0   0   0   0   0   0   0   0   0   0   0
REAL DATA CARD #   1    .6500E+02  0.         0.         0.         0.         0.         0.         0.
 COMP#.   2  INTEGER DATA     2   51   4   1  -3  -2   9   8   0   0   0   0   0   0   0   0
REAL DATA CARD #   1    .2022E+04  .2000E+04  .1500E+00  .2500E+00  0.         .1600E-01  .3026E+00  .2270E+01
REAL DATA CARD #   2    .3070E+00  .4500E+03  .8070E+02  .1430E+03  .5743E+03  .2967E+03  .3900E-01  .7000E+02
REAL DATA CARD #   3    .3000E+01  .7550E+02 -.5000E+00  .3000E-02  .7100E+01  .3000E-02  .1097E+00  .4900E+02
REAL DATA CARD #   4    .5000E+01  .4000E+04  .6000E-01  .8000E-01  .4459E-01  .1040E-01  .1000E+01  .8000E+01
 COMP#.   3  INTEGER DATA     3   61   1   2   0   0   0   0   0   0   0   0   0   0   0   0
REAL DATA CARD #   1    .1600E+05  0.         0.         0.         0.         0.         0.         0.
 COMP#.   4  INTEGER DATA     4   11   0   3  -4  -5   0   0   0   0   0   0   0   0   0   0
 COMP#.   5  INTEGER DATA     5   23   3   4  -7   0   0   0   0   0   0   0   0   4   0   0
REAL DATA CARD #   1    .2200E-01  .6500E+00  0.         0.         0.         0.         0.         0.
REAL DATA CARD #   2    0.         .1500E-01  .2400E-01  .2090E+00  0.         0.         0.         0.
REAL DATA CARD #   3    0.         0.         .1805E+01  .4805E+01  0.         0.         0.         0.
 COMP#.   6  INTEGER DATA     6   41   1   5  -6   0   0   0   0   0   0   0   0   0   0   0
REAL DATA CARD #   1    .2100E-01  .6380E+00  0.         0.         0.         0.         0.         0.
 COMP#.   7  INTEGER DATA     7   11   0   6   7  -8   0   0   0   0   0   0   0   0   0   0
 COMP#.   8  INTEGER DATA     8   41   1   8  -9   0   0   0   0   0   0   0   0   0   0   0
REAL DATA CARD #   1    .1360E+00  .6500E+00  0.         0.         0.         0.         0.         0.
 COMP#.   9  INTEGER DATA     9   61   1   9   0   0   0   0   0   0   0   0   0   0   0   0
REAL DATA CARD #   1    .6500E+02  0.         0.         0.         0.         0.         0.         0.
    CPU TIME IN SECONDS =     3.380
```

```
                              STEADY STATE INPUT DATA
        NUMBER OF NODES =  6     NUMBER OF LEGS =  6     NUMBER OF CONSTANT PRESSURE NODES =  0

                              LEG CONNECTION INPUT DATA
LEG NO      UPST NODE NO   DNST NODE NO   NO OF ELEMENTS    FLOW GUESS    UPST PRESS    DNST PRESS
[illegible]

LEG NO      ELEMENTS IN LEG----
[illegible]
```

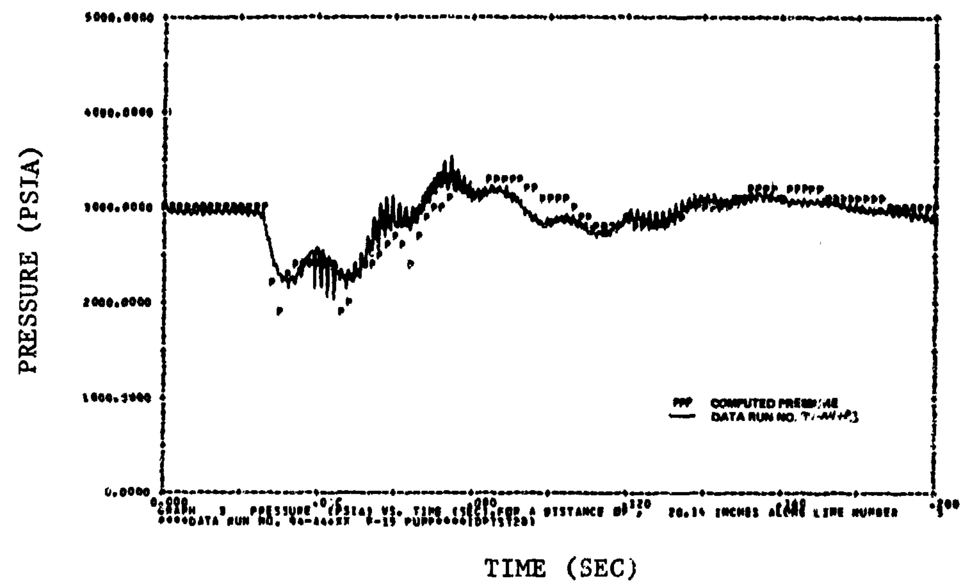


FIGURE 130 PUMP OUTLET PRESSURE RUN NO. 94-A4+P3

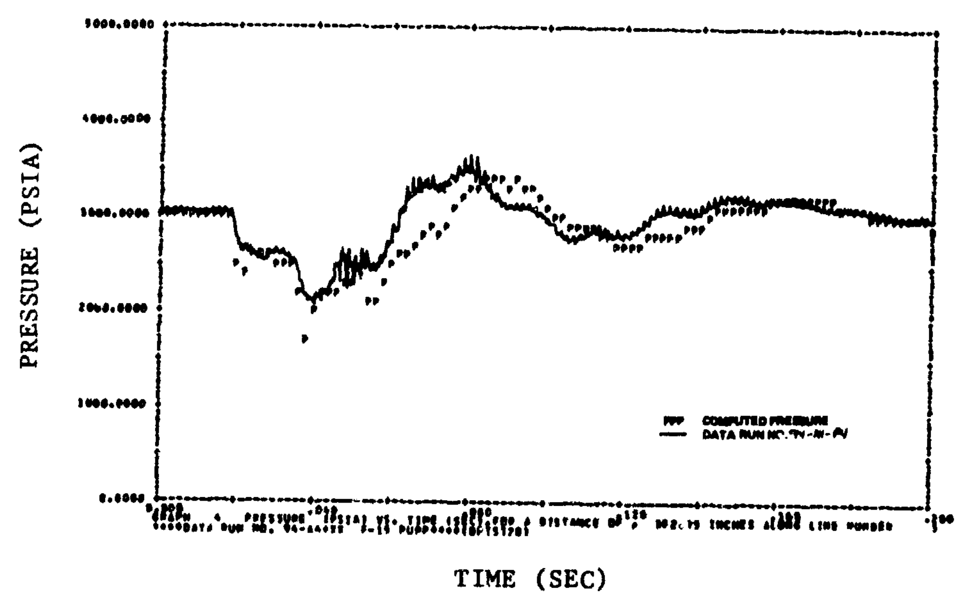


FIGURE 131 PUMP OUTLET PRESSURE RUN NO. 94-A4+P4

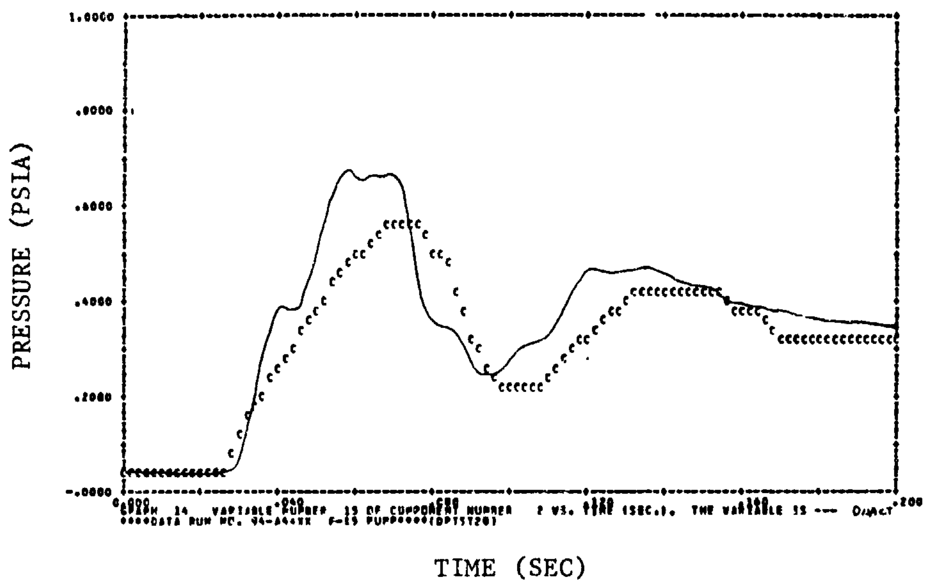


FIGURE 132 HANGER POSITION RUN NO. 94-A4+XH

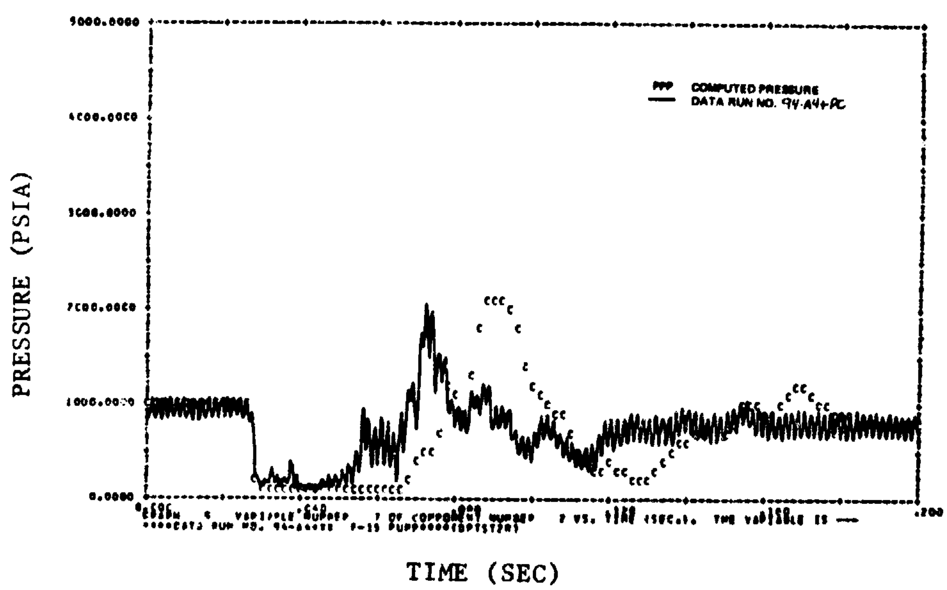


FIGURE 133 ACTUATOR PRESSURE RUN NO. 94-A4+PC

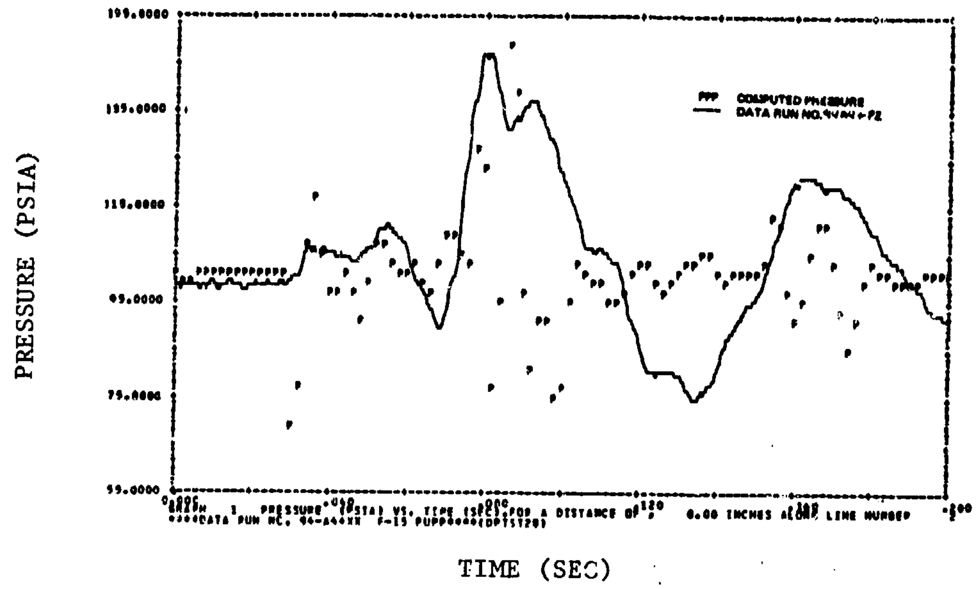


FIGURE 134 CASE DRAIN PRESSURE RUN NO 94-A4+P2

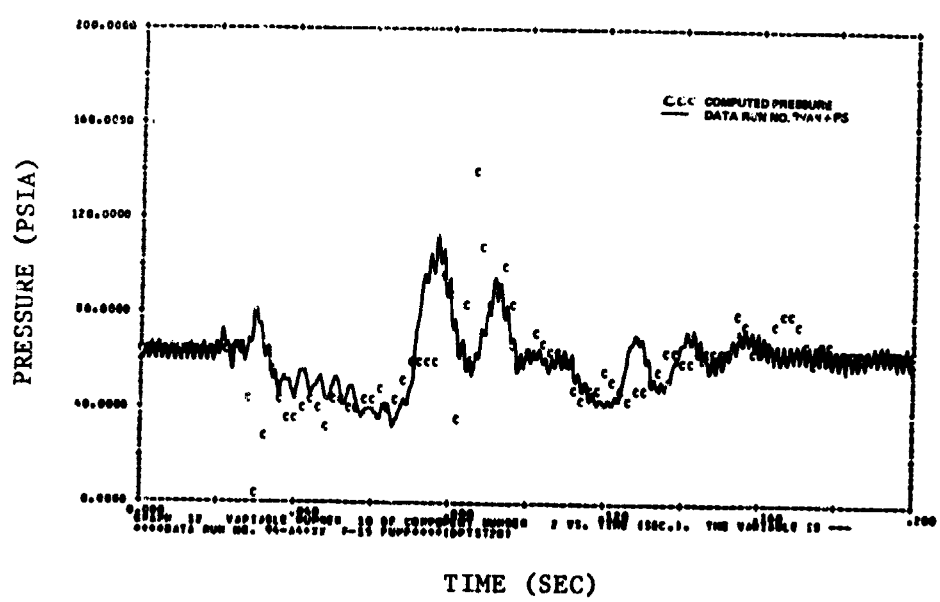


FIGURE 135 INLET PRESSURE RUN NO. 94-A4+PS

4. LOSSLESS LINE MODEL

Closely coupled hydraulic components are typical in aircraft hydraulic systems. The F-18 arresting hook actuator/damper is an excellent example. The inlet fitting to the rod side of the actuator cylinder contains a two-way restrictor. There is no interconnecting line between the restrictor and the cylinder. Similar close coupling exists with various combinations of relief valves, check valves, accumulators, and the actuator in the arresting hook unit.

A HYTRAN computer simulation of the Arresting Hook Subsystem would require either writing a single component model or using existing models to functionally represent the arresting hook unit and all the internal components. Because of design and test schedules, time is not usually available to write and verify a complete model. The functional or "building block" method is usually chosen.

The HYTRAN program requires components to be coupled with a line. To model the Arresting Hook Subsystem several lines must be added between the connected components. These lines cause errors in computing fluid friction, inertia and compressibility on transient cylinder pressures. and transient signal phasing. Therefore the feasibility of incorporating a lossless line model was investigated. A secondary objective was to reduce the user's modeling task by finding a way to couple the components without using interconnecting lines.

a. BACKGROUND

Predicting hydraulic system response is a problem in continuous system simulation. The HYTRAN program works with individual component equations, and uses the method of characteristics in the line model to connect the component models. The technique avoids algebraic equation sets and has the advantage of including frequency dependent friction effects in the predicted response.

A method of characteristics solution requires determining characteristic curves in the distance, time (x,t) plane along which the partial differential equations of momentum and energy reduce to ordinary differential equations. The ordinary differential equations can be integrated numerically, and a solution can be found which propagates through space and time.

For computational convenience the pressures and flows are computed in a set of fixed grid points. From initial conditions the solution propagates one increment in time.

The method of characteristics line model used with component models allows the computation of each component variable almost independently of other component variables. However, a line must be used to couple any two components in the system. The component algorithms in the HYTRAN program do not allow for components which cannot be coupled through a transmission line. Furthermore, the component port variables are restricted to pressure and flow.

b. HYTRAN LINE MATH MODEL

The fluid flow field is completely described by the simultaneous solutions of the momentum, continuity, energy, and state equations. For an isothermal liquid flow field the momentum and continuity equations become uncoupled from the energy and state equations. Therefore, fluid pressure and velocity can be determined from the simultaneous solution of the momentum and continuity equations alone. The equations are defined assuming one dimensional flow with the static pressure always exceeding fluid vapor pressure. The lines have a circular cross section and are full of fluid at all times. In addition, the lines are considered thin wall and all energy dissipation is due to shearing stresses at the walls.

The transmission line model defines the response in terms of the equation of motion,

$$\frac{a}{\rho a} \frac{\partial P}{\partial x} + \left(\frac{\partial V}{\partial t} + V\frac{\partial V}{\partial x}\right) + F(V,D) = 0 \tag{50}$$

and the continuity equation.

$$\frac{1}{\rho a} \frac{\partial P}{\partial t} + \frac{1}{\rho a} V \frac{\partial P}{\partial x} + a \frac{\partial V}{\partial x} = 0 \tag{51}$$

Where

P = Pressure (psia)

V = Fluid Velocity (in/sec)

t = Time (sec)

ρ = Fluid Mass Density $\left(\frac{\text{lb-sec}^2}{\text{in}^4}\right)$

$F(V,D)$ = Pressure loss as a function of velocity and pipe diameter

a = Speed of sound in fluid (in/sec)

x = Distance along line (in)

The method of characteristics allows equations (50) and (51) to be transformed into a pair of total differential equations. The equations are valid only when restricted to characteristic lines in the x-t plane (Figure 136).

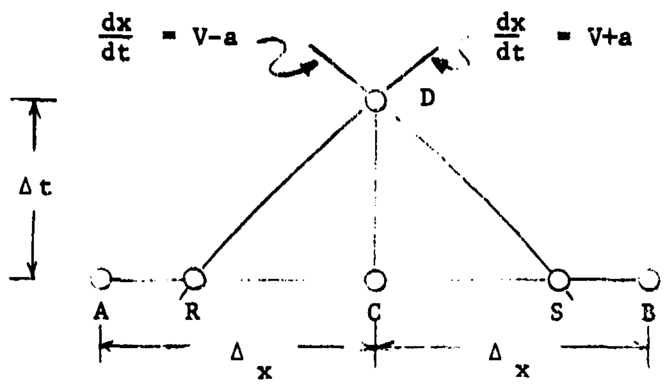


FIGURE 136. CHARACTERISTICS IN X-T PLANE

Taking the sum and difference of equations (50) and (51) and using the definition of the total time derivative,

$$\frac{d}{dt} = \frac{\partial}{\partial x}\Big)_t \frac{\partial x}{\partial t} + \frac{\partial}{\partial x}\Big)_x$$

and assuming $Z = \frac{\rho a}{A}$ (the characteristic impedance of a frictionless line)

one can write, making the appropriate substitutions,

$$\frac{dP}{dt} + Z\frac{dQ}{dt} + F(Q) = 0 \qquad (52)$$

which is valid along the curve

$$\frac{dx}{dt} = V + a \qquad (53)$$

and

$$-\frac{dP}{dt} + Z\frac{dQ}{dt} + F(Q) = 0 \qquad (54)$$

valid along the curve

$$\frac{dx}{dt} = V - a \qquad (55)$$

Where F (Q) = pressure loss as a function of fluid flow (psi).

The original partial differential equations (50) and (51) have been replaced by the two ordinary differential equations (52) and (54). These equations are only valid along the time space curves defined by equations (53) and (55) which are called the left and right characteristics respectively.

The pressure and velocity at the points R and S in Figure 1 can be interpolated since all the quantities at A, B, and C are known. Assuming that the wave characteristics are straight lines from R to D and S to D a first order integration of equations (52) and (54) results in a set of simultaneous difference equations. The simultaneous solution for P_D and Q_D results in

$$P_D = \frac{C_R + C_L}{2} \tag{56}$$

$$Q_D = \frac{C_L - C_R}{2Z} \tag{57}$$

Where

$$C_L = P_R + Z * Q_R - F_1(Q_R) - F_1(Q_D)$$

$$C_R = P_S - Z * Q_S + F_2(Q_S) + F_2(Q_D)$$

C_R and C_L are still functions of the unknown flow, Q_D. In HYTRAN an iteration is used to compute Q_D. C_R and C_L are first determined using only a rectangular rule to approximate the friction. This gives a very close approximation of Q_D which is then improved by using the trapezoidal rule to evaluate C_R and C_L.

There are two ways to apply these equations to a line system. Either modify the velocity of sound slightly to make the number of calculation grids in each individual line an integer such that

$$n = \frac{L}{a * \Delta t} + 1 \tag{58}$$

where

n = number of grids (an integer value)

L = line length (IN)

a = wave speed (in/sec)

Δt = calculation interval (sec)

or use a grid of characteristics. The method of specified time intervals, which modifies the wave speed, is used in HYTRAN. This offers convenience in computer simulation because x_D and t_D are assigned definite values, only P_D and Q_D are unknown.

If either Q_D or the wave speed vary considerably with x and t, as for highly deformable tubes, then the grid of characteristics is preferred.

By using the interpolation method outlined above, it is possible to just satisfy the limitation on the grid-mesh ratio in each line and proceed with the solution. However, with linear interpolation on the x grid, the larger the interpolation, the less accurate the numerical solution. The Δt should be chosen so computing time is not excessive. The Δx selection will insure that interpolations are applied over minimum distances in each line.

For the method of characteristics solution to be stable the points R and S must lie between A and B. For the maximum magnitude of V+a one must assume

$$\Delta t \leq \frac{\Delta x}{|V+a|} \tag{59}$$

The fluid velocity is typically much smaller than sonic velocity in a hydraulic system. Therefore, the grid size (Δx) implicitly determines the allowable step size (Δt). The converse is also true. Typically, the smaller grid size produces a better definition of the pressure and flow transients. Larger step sizes reduce the accuracy of the model. The most simple approach to achieve more accurate results is simply to use a smaller time step and grid size.

However, for large system simulations, this becomes impractical. The length and cost of the computer simulation is often prohibitive.

Equations (56) and (57) apply only to the interior sections of the fluid line. The propagation of all quantities along the interior spatial grid lines depend only on the values along the current time grid line. Therefore, all interior point quantities can be propagated one time increment independent of the boundary conditions which are imposed by the components on each line end.

The grid points on the boundaries of a line cannot be propagated independent of the connected component. The pressure and flow at the line ends are indeterminate because one wave characteristic is not present. An algebraic equation in terms of the wave characteristics must be combined with the independent port variable to obtain the solution for pressure and flow at the line endpoints.

c. LOSSLESS LINE MODEL DEVELOPMENT

Various programming and analytical ideas were studied to provide a lossless line model as described in the introduction. These have been categorized into three approaches, the practical, direct coupling, and port energy method.

(1) PRACTICAL APPROACH

The method of characteristics in the HYTRAN line model solves for pressure and flow using specified time intervals. The wave speed in the line is determined from fluid properties with corrections for pipe wall elasticity. The calculation time interval is input by the program user. The minimum calculation interval for line distance $(\overline{\Delta x})$ is therefore specified for each computer run as,

$$\overline{\Delta x} = \text{wave speed} * \text{time interval} \qquad (60)$$

With an integration step of 0.0001 seconds and an acoustic velocity of 50,000 in/sec, the minimum Δx line segment would be 5.0 inches. A line with five segments would be 25 inches long. If the time increment was reduced, the number of Δx's would increase and shorter lines could be more accurately modeled. The reductior in times step will increase the computer running time for the same realtime simulatior

For calculations of pressure and flow at the characteristic grid points during the same time step, the number of Δx's in the line are determined by equation (58). The Δx used in the simulation is computed as

$$\Delta x = \frac{\text{line length}}{n} \qquad (61)$$

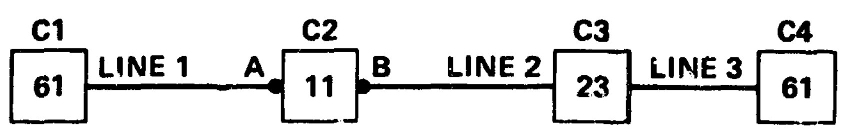
Depending on the time interval selected, an error may occur between $\overline{\Delta x}$ and Δx. (Note: n is always an integer.) A linear interpolation is used in HYTRAN to obtain the interior values of pressure and flows at point R and S in Figure 1 from the known values at A, B and C.

If the Δx is more than the line length between closely coupled components, the method of specified time intervals requires reducing the velocity of sound in the line so as to make the line length an exact multiple of Δx. During the transient simulation, line dynamic friction effects are omitted and the line will be one Δx in length with no intermediate calculation points. The right and left characteristics are determined using the smallest Δx in the simulation. The short line segment's characteristic impedance and static friction losses are based on the actual line length and diameter. Since the velocity of sound and fluid density are constant in the HYTRAN program, the characteristic impedance is affected only by the line's cross-sectional area.

If the user selects a grid size greater than the smallest line length, the simulation error results from the fix-up taken in the acoustic velocity. The error can be minimized by a proper selection of the line's characteristic impedance and elimination of the static friction losses. The friction losses can be easily deleted, and the phasing errors due to the fix-up in acoustic velocity can be affected by the line's cross-sectional area.

An inaccurate diameter for a line smaller than the grid Δx can significantly affect the computer simulation. To illustrate this, several computer runs were made.

A simple baseline system (Figure 137) with two constant pressure reservoirs, a branch, three lines, and a two-way, two position valve was chosen for the simulation. In the HYTRAN program the branch model has no static or dynamic friction losses. Consequently, the pressures and flows are the same at points A and B in Figure 137 . The HYTRAN input data for the simulation is presented in Table 20.



C1, C4 = CONSTANT PRESSURE RESERVOIRS
C2 = BRANCH
C3 = TWO-WAY, TWO POSITION VALVE
LINE 1, 2, 3 = 1/2" O.D. X .020" WALL X 100" LONG

FIGURE 137. BASELINE LINE SYSTEM

TABLE 20. HYTRAN INPUT DATA FOR BASELINE SYSTEM

**** LINE SYSTEM NO LOSSLESS LINE SEGMENT - TURN OFF (DLOSS2A)

	.0001		.1		.001		100.					
3	4	1										
1	0		0						100.	.5	.028	3.0E7
2	0		0						100.	.5	.028	3.0E7
3	0		0						100.	.5	.028	3.0E7
1	61	1	-1									
	3000.											
2	11	0	1	-2								
3	23	3	2	-3							4	
	0.10		.65									
	0.		.010		.012		.5					
	.1		.1		0.		0.					
4	61	1	3									
	100.											
3	2											
1	1	2	5		1.							
1	1	0	1	2	1	0	2	3	1			
2	2	3	2		1.							
0	3	4	1									
2	0		1									
1	2		100.		-100.							
2	4		.1		-.1		100.		-100.			

A branch and a short line segment less than a Δx length was added to the circuit in Figure 132. Line 2 in Figure 138 is the short line segment. Points A and B in Figure 138 correspond to the same points in Figure 137.

To minimize the effects of the short line segment a lossless line model was added to the line subroutine. Table 21 contains a listing of the HYTRAN input data. The lossless line model is denoted by a line type 2.

The lossless line model has no static or dynamic friction. The velocity of sound in the pipe is determined as

$$a = \sqrt{\frac{\beta}{\rho}} \qquad (62)$$

where β = fluid bulk modulus (Psi)

Equation (62) assumes the acoustic velocity is not affected by the line wall thickness and the material modulus of elasticity. Since friction effects are not included in the model, the length of the lossless line segment is immaterial. A computer run was made with the system in Figure 138. The lossless line length was 0.1 inch. The I.D. of all the lines were identical. The effect of the lossless line is shown by comparing pressure response at point A. Similar comparisons could be made at point B or any other point in system. Waveform would be the same at other locations, but pressure levels would vary slightly due to steady state losses at locations other than point A. An overplot of the baseline pressure at point A in Figure 137 and the lossless line results at point A in Figure 138 is shown in Figure 139. Initial steady state flow was 53.0 CIS before the valve was closed. There is a slight mis-phasing between the baseline and lossless line pressure curves. Other computer runs were made using a 0.078", 0.884" and 3.5" inner line diameters for the lossless line. The results of the simulations are plotted for point A in Figures 140, 141, and 142.

Compared to the baseline run at Point A the .078" lossless line has slightly attenuated the pressure signal and decreased the damping frequency. The .884" diameter run was closer to the baseline but the largest diameter in Figure 142 completely changed the wave shape from the baseline. Since the velocity of sound in the lossless line was the same for each run, the differences are due to cross-sectional area regardless of the lossless line segment length, as long as the length is less than the smallest calculation grid. The proper selection of the line diameter is important when using the lossless line model.

Another set of runs were made using a 1/4" OD x 100" long line downstream of the tee in Figure 137. A lossless line segment with a 0.444" ID was put in the system and the simulation results compared to the baseline system pressure at point A is shown in Figure 143. There are minor phasing errors between the baseline and lossless case, but amplitudes still show good correlation. A computer run was tried using a 0.206" ID lossless line segment. The results of the simulation for pressure at point A is shown in Figure 144, and is essentially the same as Figure 143.

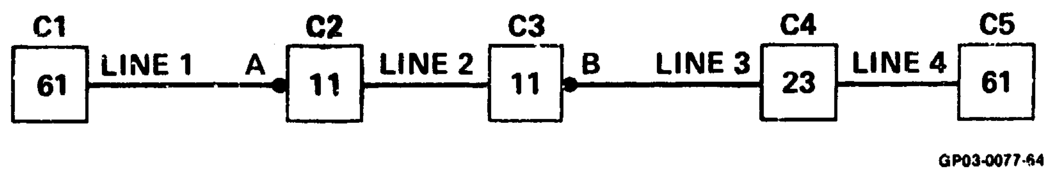


FIGURE 138. LINE SYSTEM WITH LOSSLESS LINE SEGMENT

TABLE 21. HYTRAN INPUT DATA FOR LOSSLESS LINE SYSTEM

```
**** LINE SYSTEM WITH LOSSLESS LINE SEGMENT - TURN OFF (DLOSS2)
   .0001           .1       .001       100.
  4    5    1
  1    0         0                              100.       .5      .028     3.0E7
  2    2         0                              0.1        .5      .028     3.0E7
  3    0         0                              100.       .5      .028     3.0E7
  4    0         0                              100.       .5      .028     3.0E7
  1   61    1   -1
   3000.
  2   11    0    1   -2
  3   11    0    2   -3
  4   23    3    3   -4                                                  4
     .10       .65
      0.      .010      .012       .5
      .1        .1        0.       0.
  5   61    1    4
    100.
  3    2
  1    1    2    7         1.
  1    1    0    1    2    1    0    2    3    1    0    3    4    1
  2    2    3    2         1.
  0    4    5    1
  3    0         1
  1    2      100.     -100.
  2    4       .01      -.01        .1      - .1
  3    4        .1       -.1      100.     -100.
```

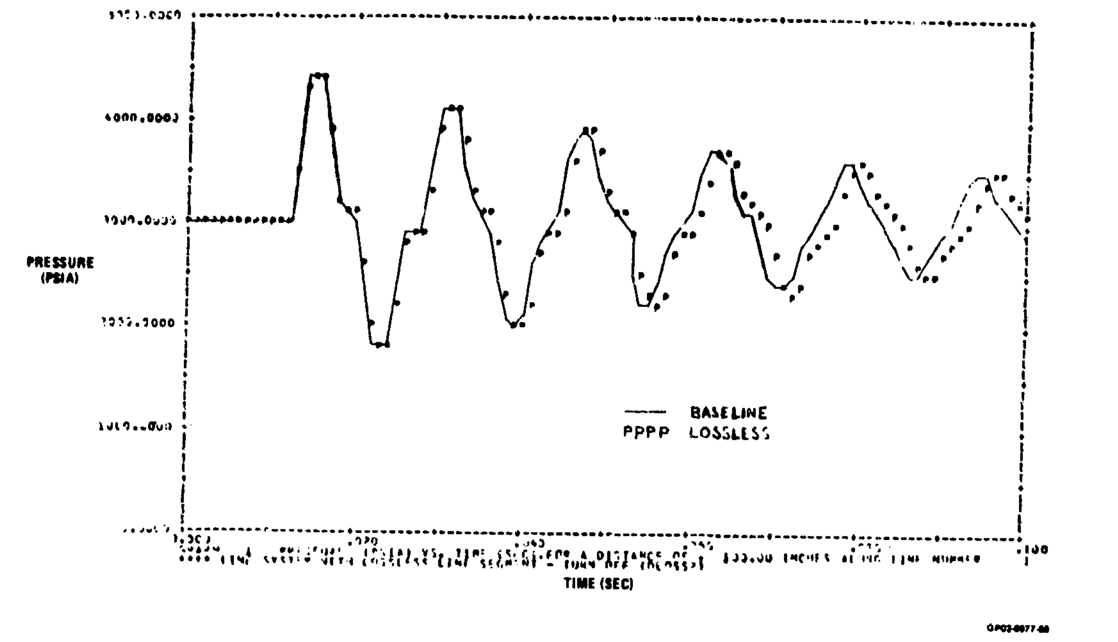


FIGURE 139. **COMPARISON OF A 0.444 IN. ID LOSSLESS LINE WITH A 1/2 IN. LINE SYSTEM 100 IN. ALONG LINE 1**

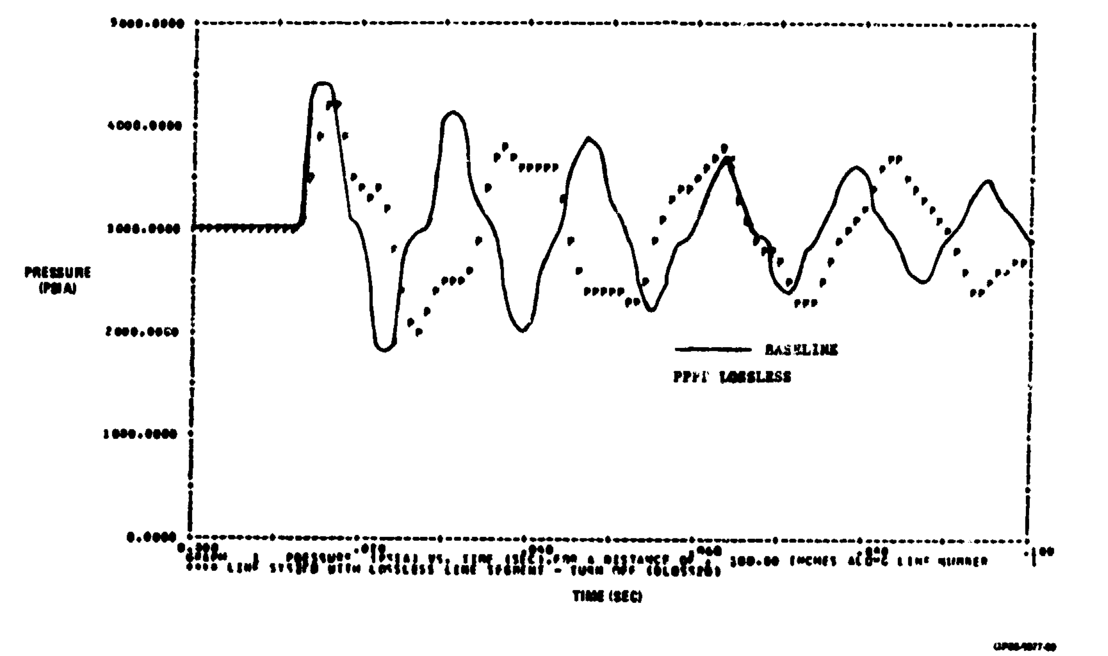


FIGURE 140. **COMPARISON OF A 0.078 IN. ID LOSSLESS LINE WITH A BASELINE 1/2 IN. LINE SYSTEM 100 IN. ALONG LINE 1**

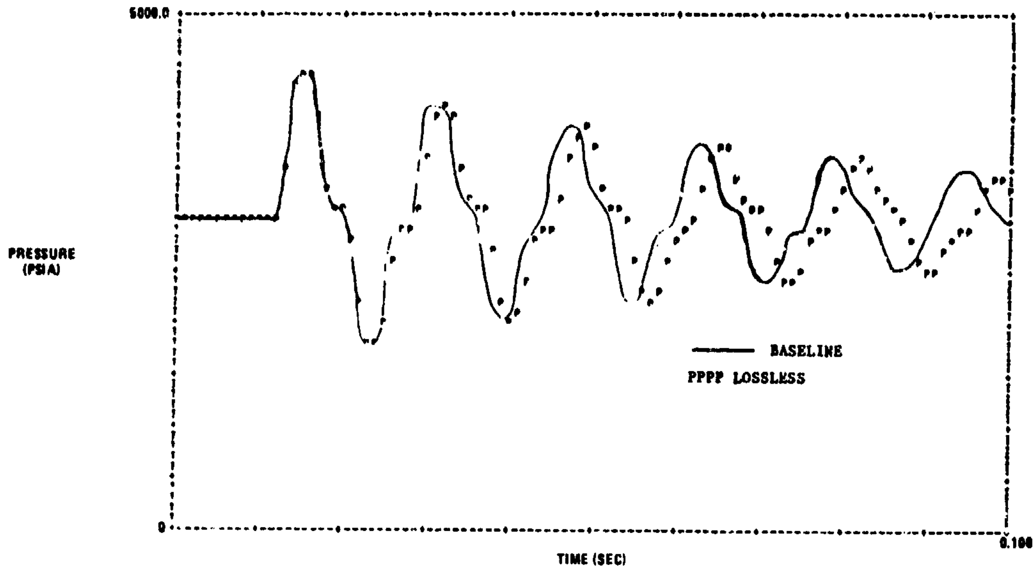


FIGURE 141. COMPARISON OF A 0.884 IN. ID LOSSLESS LINE WITH A BASELINE 1/2 IN. LINE SYSTEM 100 IN. ALONG LINE 1

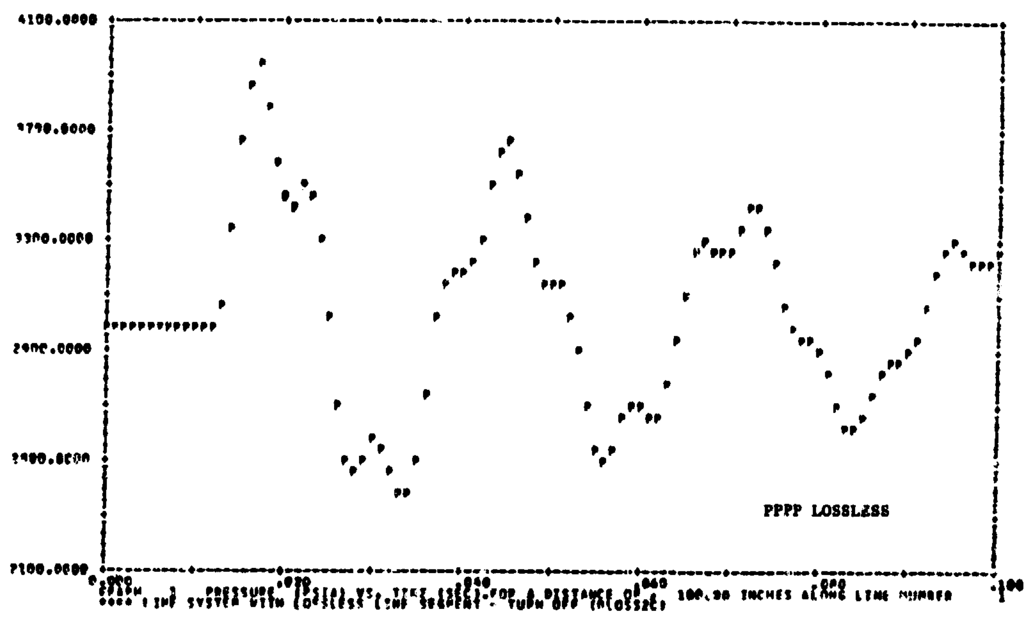


FIGURE 142. 3.5" ID LOSSLESS LINE COMPUTER RUN

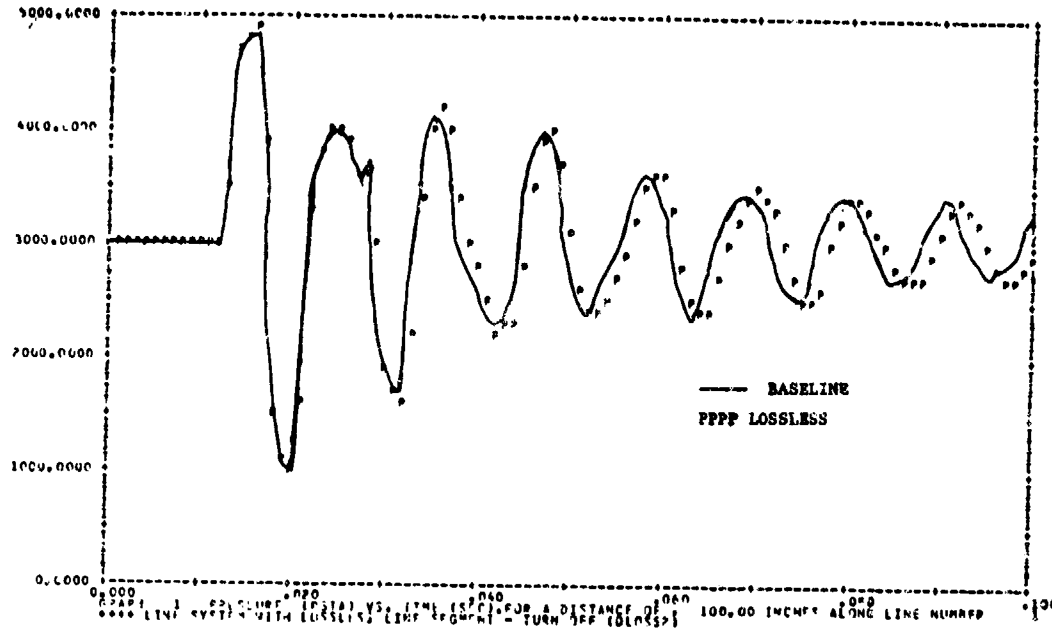


FIGURE 143. COMPARISON OF A 0.444" ID LOSSLESS LINE WITH A BASELINE 1/2" and 1/4" LINE SYSTEM

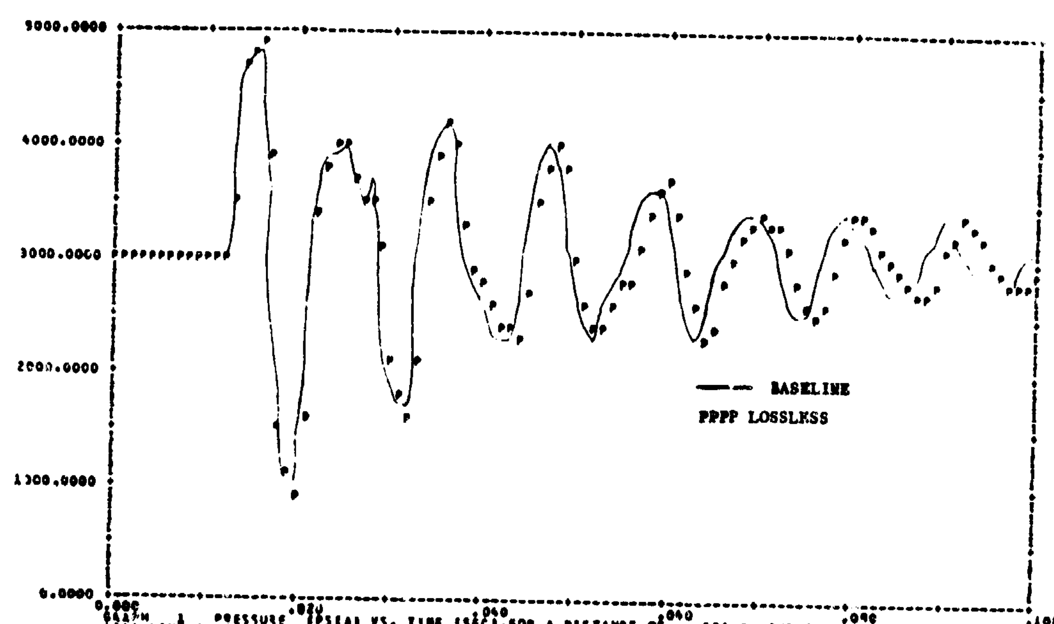


FIGURE 144. COMPARISON OF A 0.206" ID LOSSLESS LINE WITH A BASELINE 1/2" and 1/4" LINE SYSTEM

A run was made substituting a 4.0" ID x 10" long line for line 2 in the baseline system in Figure 137. The base line pressure at point A is compared to the lossless line runs using a 0.444" ID and 4.0" ID in Figures 145 and 146 respectively. Using the 0.444" ID line gives a better approximation than the 4.0" ID line.

Figure 147 shows the effect of the lossless line model when the line length is greater than the minimum calculation grid (Δx) size. There are two Δx's in the lossless line. The lossless line had a 0.444" ID and the system line lengths were those shown in Table 21. The frictionless case has slowed the waveform because of the delay in traveling across the two Δx segments.

(2) DIRECT COUPLING APPROACH

Direct component coupling using the method of characteristics for transporting fluid flow and pressure involves two techniques. In the first, the HYTRAN program would create frictionless lines between the connected components. The major stumbling block is the proper selection of a line area by the program. Most of the HYTRAN component models do not have port diameters. If they did, the area's would not necessarily be the correct ones to use. What area would the program select for the Arresting Hook Subsystem's two-way restrictor on the inlet port of the actuator? Even though the programming would be relatively straight-forward, selecting an appropriate port diameter is best left to the program user because of the varying design conditions encountered. A lossless line model would be better than this technique.

The second technique would omit the closely coupled line. The method of characteristics requires the simultaneous solution of the line end boundary conditions with the component equations for the pressures, flows, and other component variables. With the line segment removed the components are directly coupled and the component algorithms must be solved simultaneously for the pressure and flow upstream and downstream of the connected components as well as the component variables.

This would be a large programming task. Every component subroutine in HYTRAN would require the addition of a submodel. The submodels could be linked through appropriate input flags on the component integer cards. New control subroutines would be required to combine the submodels and perform the computations. The size of the HYTRAN program would probably double in core size and the running time would significantly increase even for the simplest systems. The complexity of this approach eliminates any advantages obtained from reducing the number of lines. It is impossible for this technique to be any more efficient than the lossless line approach, or decreasing the time step size for a more accurate simulation.

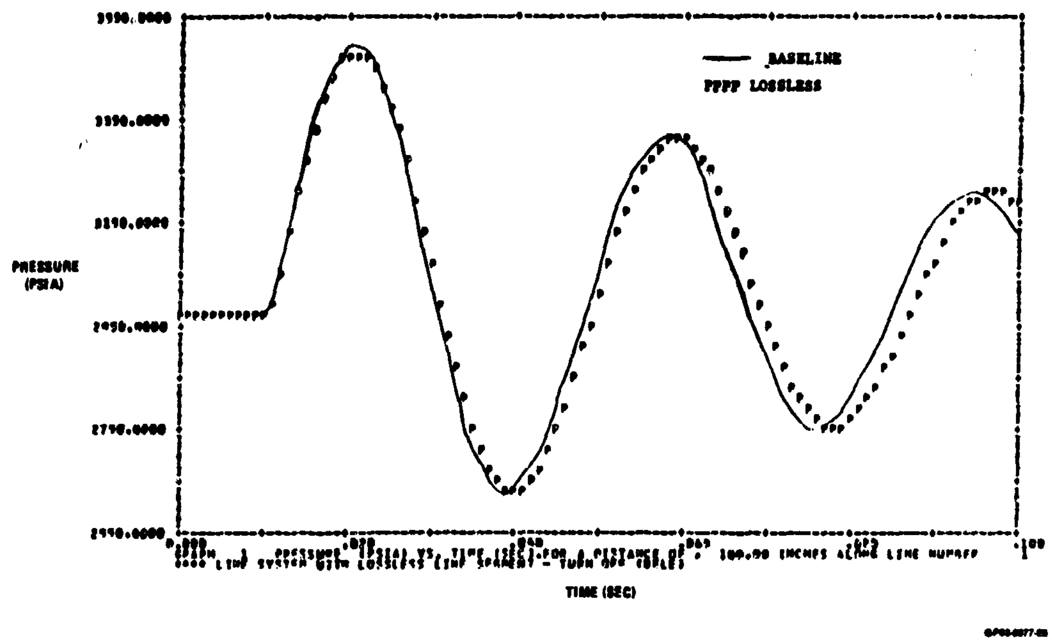


FIGURE 145. **COMPARISON OF A 0.444 IN. ID LOSSESS LINE WITH A BASELINE 1/2 IN. AND 4.5 IN. LINE SYSTEM 100 IN. ALONG LINE 1**

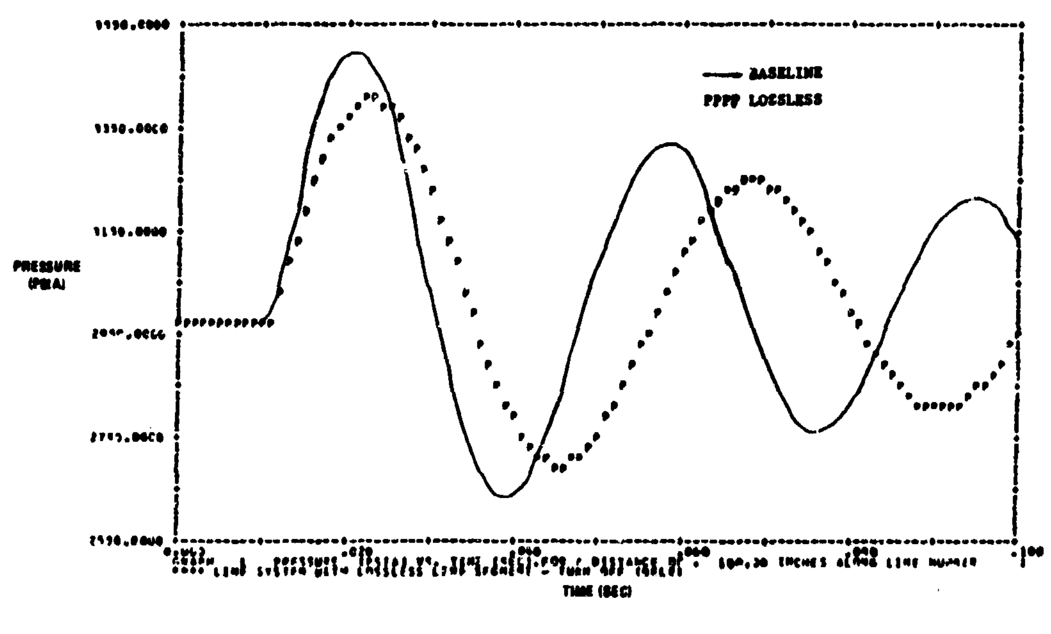


FIGURE 146. **COMPARISON OF A 4.0 IN. ID LOSSLESS LINE WITH A BASELINE 1/2 IN. AND 4.5 IN. LINE SYSTEM 100 IN. ALONG LINE 1**

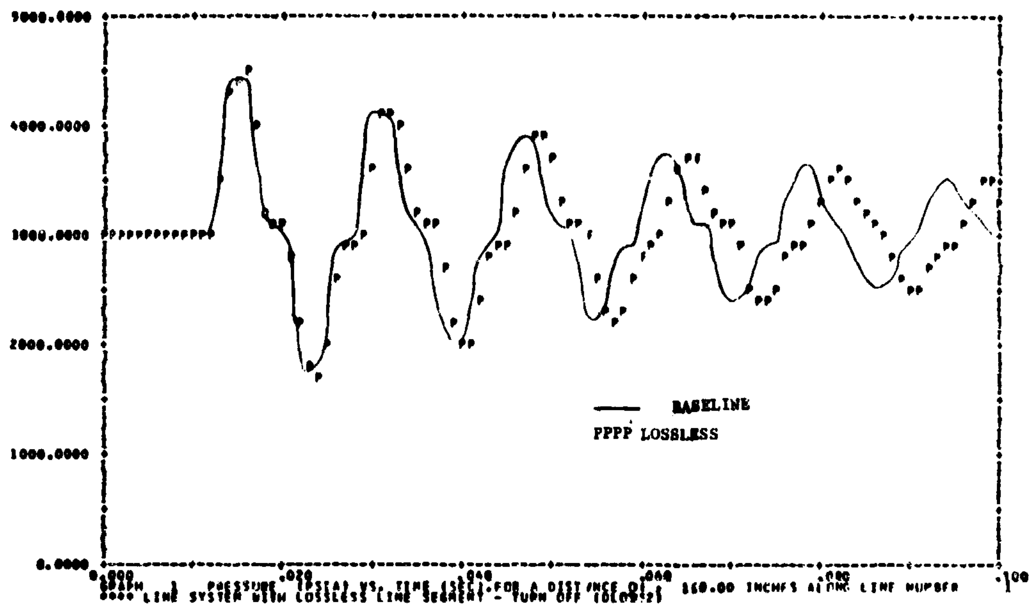


FIGURE 147. RUN WITH TWO DELTA X'S IN A 0.444" ID LOSSLESS LINE

(3) PORT ENERGY METHOD

Analytically the best approach would be to develop port energy models for all the components in the HYTRAN program.

Development of a block-oriented simulation program would require components to be modeled in terms of the physical variables which are defined at the component ports. These variables are called port variables where "port" refers to the points at which components are interconnected.

Port variables for multiport component models can be chosen in two ways. One method defines port variables such that the product of the two variables at a port equals the rate at which energy passes through the port. In the other method, the product of the port variables represents the net energy which has crossed the port.

Hydro-mechanical systems can be modeled in terms of five port variable pairs which are (pressure, flow), (force, position), (force, velocity), (torque, angular position), and (torque, angular velocity). Thermal effects could be included by two additional port types which include the effect of heat transfer and the internal energy in a fluid stream and a wall mass.

The ports are the independent and dependent variables of the component model. The coupling of component models to form a system model causes the independent and dependent variables at connected ports to be interrelated. Therefore, a port can be independent with respect to one component, dependent with respect to the connected component, and unknown with respect to the system model. Equations are established which allow for the simultaneous solution of the system model parameters.

The simulation can be considered as a type of network analysis. The advantage over an electrical network simulation is the ability to include the non-linear response of the hydraulic components and the use of a multi-port model which usually cannot be decoupled from the system. Each port can be connected to the appropriate port of another component. Once the system is assembled all the port variables are solved simultaneously.

Algorithms developed by Smith (REF (11)) and Ebbesen (REF (12)) assume a system can be represented as a set of coupled components. Each component is described by differential, algebraic, and difference equations, and port variables provide the inputs and outputs. There are no restrictions concerning acceptable ports as long as two variables exist at each port with one being independent and the other dependent with respect to a reference component.

The port energy method would require the complete rewriting of the HYTRAN program. Although the approach allows for direct connection of components and could reduce the amount of computer core storage and program execution time, the manpower time and cost of implementing these changes are prohibitive within the present contract.

5. HYTRAN PROGRAM IMPROVEMENTS

The HYTRAN program improvements consist of six new or improved component models. These improvements will allow simpler data input in some cases and generally provide more accurate component simulation.

In addition to component models several enhancements were made to the utility subroutines. The improvements are discussed in the following sections.

a. SUBROUTINE VALV21

The simple two way control valve model (Subroutine VALV21) has been changed to allow the input data to consist of rated flow and pressure drop or the slot width and discharge coefficient as required before. This allows the user to simulate a valve without having detailed knowledge of its internal workings. See Appendix E for the user and technical manual sections.

b. SUBROUTINE CVAL31

The dynamic solution technique of the check valve model (Subroutine CVAL31) has been changed to allow a more stable simulation. The improved model stability will reduce any system instabilities caused by an oscillating poppet, as was common with the old model. The model can also be used to simulate high cracking pressure relief valves. Appendix F contains the user and technical manual sections.

c. SUBROUTINE CREL32

The priority valve model (Subroutine CREL32) has been updated to be more representative of this type of component than the old model was. As before, it is still a simple model, simulating flow characteristics instead of spring/mass/flow relationships. The subroutine cannot be used as a detailed dynamic model of a priority valve, but does provide a useful component in total system simulation. The user and technical manual write-ups are in Appendix G.

d. SUBROUTINE REST41

The two way restrictor model (Subroutine REST41) has been changed to allow the user to model its characteristics by either rated flow and pressure drop or orifice diameter and discharge coefficient. Cavitation models have been added to both ports. See Appendix H for the user and technical manual sections.

e. SUBROUTINE RSVR63

A level sensing (RLS) bootstrap reservoir model (Subroutine RSVR63) has been added. The model combines the characteristics of four different components (one branch, one bootstrap reservoir, and two restrictors) into one. Appendix I contains the user and technical manual write-ups.

f. SUBROUTINE ACT108

A dual system, tandem, valve controlled actuator (Subroutine ACT108) simulates a two system, linear, valve controlled actuator of the type commonly used in flight control actuation systems. The model provides the mechanical interface between two separate hydraulic systems. The user and technical manual sections are in Appendix J.

g. SYSTEM ARRANGEMENT DATA

An explantion of the zero flow leg feature has been added to the arrangement data section of the HYTRAN user manual. The revised section is shown in Appendix K.

h. OUTPUT DATA REQUIREMENTS

The user can select the Y axis scales for any output graph. The required user information is presented in Appendix L.

i. OUTPUT SUBROUTINES

Changes made to the technical manuals output subroutines are documented in Appendix M.

j. FUNCTION SECORD

The function secord provides a second order response to either a step input or impulse forcing function. The function is used by the empirical pump model and the HYTRAN technical manual write-up is in Appendix N.

k. BLOCK DATA AND COMP SUBROUTINE

The block data and comp subroutine listings incorporating the new models is presented in Appendix O.

l. LINE MODEL AND DYNAMIC FRICTION

The lossless line model changes are documented in Appendix P.

SECTION III
HSFR PROGRAM

The Hydraulic System Frequency Response program (HSFR) was investigated with the objective of improving the accuracy of predicted pulsation amplitudes, developing techniques for empirically determining the internal impedance of a typical pump, and to further extend program capabilities. A key element of the investigation of empirical techniques was to have been development of an oscillating piston acoustic source. However, mechanical problems with bearings in the device precluded gathering meaningful data. Program revisions were made to improve the flow calculation in the pump model. The main program was revised to include the capability to plot the standing pressure wave in a given section of the circuit at the option of the user.

1. BACKGROUND

The HSFR program was developed to simulate the dynamic response of a hydraulic system to the acoustic noise generated by the pump. The program predicts how oscillatory flows and pressures caused by the acoustical energy content of a pump output are transmitted through lines and components of a hydraulic system. Resonance can occur if a frequency of the oscillatory output coincides with a natural frequency of the system. Resonant conditions can produce large oscillatory pressure and flow amplitudes which, in turn, can cause excessive line motion and stresses, resulting in premature failure of system lines and components.

The HSFR program predicts the pump speeds at which major resonances occur with excellent correlation accuracy. However, the prediction of peak pressure pulsation amplitude has typically been significantly inaccurate; i.e., up to 30% or greater above measured values.

The capabilities of available instrumentation provide good measurement of pressure values. However, flow measuring devices have been unable to provide sufficiently accurate data to allow intelligent modification of the program based on the results of empirically determined definition of the dynamics of the interaction between the pump and system. Accordingly, the major thrust of

current investigative efforts concentrated on the definition and calculation of the dynamic flow signal which is output by the pump.

2. AMPLITUDE PREDICTION

The amplitude of pressure pulsations in the circuit depend on the relationship between the dynamic flow signal produced by the pump and the impedance of the circuit load. The program calculates the dynamic flow output of the pump taking into account the variation of swash angle with RPM required to supply steady state and leakage flow demand, the pump valving geometry, and characteristics of the fluid being used.

The pump is modeled as an acoustic source with a parallel shunt impedance. The dynamic flow signal generated by the pumping action of the pistons is modified by the pump internal impedance before traveling to the outlet of the pump where interaction with the circuit load (impedance) produces dynamic pressure pulsations.

a. Study Results

The PUMP subroutine of the HSFR program was reviewed. The swash angle calculation was completely revised to provide better steady state flow balance. The flow calculation was re-derived and routines were included to insure calculation stability. Definition of the dynamic flow waveform by Fourier Analysis of time dependent outlet flow remained essentially unchanged. The dynamic pressure-flow balance section was completely revised to analyze the internal (shunt) dynamic flow. Inlet flow calculation and inlet pressure-flow dynamic balance sections of the program were revised to incorporate the changes made for outlet analysis. The torque calculation section of the program was not changed.

A technical description and listing of the revised PUMP subroutine is included in Appendix A.

(1) Valve Area Calculation - Equations for an exact solution to the area calculation for the piston slot opening to the valve plate pressure and suction slots were developed. The previous math model made certain simplifying assumptions relative to the geometry of the valve openings which resulted in approximation errors in the calculation. The errors introduced by these approximations were small, but tended to be cumulative in nature. The revised program eliminates the approximation errors.

(2) Steady State Swash Angle Calculation - This section was revised to provide an exact balance (within ± 0.01 CIS) between input and calculated steady state flow demand. The previous program assumed that balance was achieved when the calculated steady state flow error fell below + 0.01 CIS. However, analysis of the results obtained showed that, after the swash angle calculation for the first several RPM's, the error would overshoot in the negative direction. This resulted because the swash angle for each succeeding RPM was based on the previously calculated value. The incremental swash angle used to make the adjustment was based on an estimate of the incremental flow change per RPM. This method proved to be in error.

The program was revised to calculate the swash angle for each RPM independently. An interpolation routine between minimum and maximum flow capability at each RPM is used. The revised program includes an iteration balance routine so that undershoot error is recognized and corrected.

(3) Flow Calculation - The equations defining output for one piston of the pump as it traverses the pressure slot were re-derived using flow balance criteria. The previous program used pressure balance criteria to establish time dependent flow. This approach is subject to sensitivity error because very small changes to quite large pressure values are used to predict flow.

The revised calculations provide a check and balance iteration between the results of two independent equations to predict flow. The iteration process continues until an absolute error less than 1/2% between succeeding flow calculations is achieved. The calculation converges rapidly on flow balance, usually requiring no more than four iterations.

(4) Dynamic Pressure-Flow Balance - The most significant changes to the PUMP subroutine were made in this section. The previous program estimated a value for pump internal shunt impedance and used the resulting total impedance to produce a complex dynamic test pressure. Flow calculations were repeated and the resulting difference in output flow was used to correct the internal impedance and dynamic pressure values. The third calculation of pump flow was assumed to be the flow passed to the circuit, from which dynamic pressure was derived.

The revised program uses the first calculation of flow (which assumes zero dynamic pressure and therefore zero internal impedance) with circuit

impedance to produce a test pressure. Flow is recalculated, as before, except that an exact value of internal impedance is calculated using the known dynamic pressure and pump total flow values. The exact value of internal impedance is then used to calculate shunt dynamic flow.

A new test pressure is produced equal to 1/2 of the previous value. Flow is again calculated, as are shunt impedance and flow for the new test pressure.

The key to this approach is that exact shunt impedance is calculated for each case. There are no estimations involved and no corrections, adjustments or approximations are introduced. Shunt impedance for the two test pressures are nearly identical; the differences between them may be accounted for in the Fourier Analysis of total pump output flow.

Values are now known for shunt impedance and shunt flow for the same circuit (pump model) at different conditions (two test pressures). The Norton Equivalent circuit electrical analogy may be used to calculate output dynamic pressure. This dynamic pressure may now be used with the circuit load impedance to determine that part of total pump dynamic output flow which passes to the circuit.

(5) Other Changes - The inlet flow calculation and inlet dynamic pressure-flow balance sections of PUMP were revised to incorporate the changes made in the outlet flow calculation and outlet dynamic pressure-flow balance sections, respectively. Only minor changes were made in the remaining sections.

Preprogrammed write options were included at appropriate places in the PUMP subroutine so that selected data calculated by the program may be written for detailed analysis of pump performance.

b. Verification Tests

The primary verification of the changes made to the program was by comparison to previously conducted F-15 pump verification tests. These tests were conducted as part of the original program development effort and are discussed in Section IV of Reference (1). New verification tests using a similar short line test circuit with an F-15 pump were also conducted.

(1) F-15 Pump Verification Test - The 9 ft. test circuit shown in Figure 47 of Reference (1) was modeled for use with the revised program. Three conditions for which original data is available were run:

- Run 64-83-P1 (0 GPM) Figure 51, Reference (1)
- Run 64-81-P1 (2.0 GPM) Figure 53, Reference (1)
- Run 64-80-P1 (10.0 GPM) Figure 54, Reference (1)

The revised program provides significantly improved amplitude correlation with the recorded data as shown in Figure 148 except for the 10 GPM flow case. Analysis of the trends of both computed and measured data suggest that the peak pressure for 10 GPM at the 3850 RPM resonance should be in the range of 450-500 psip.

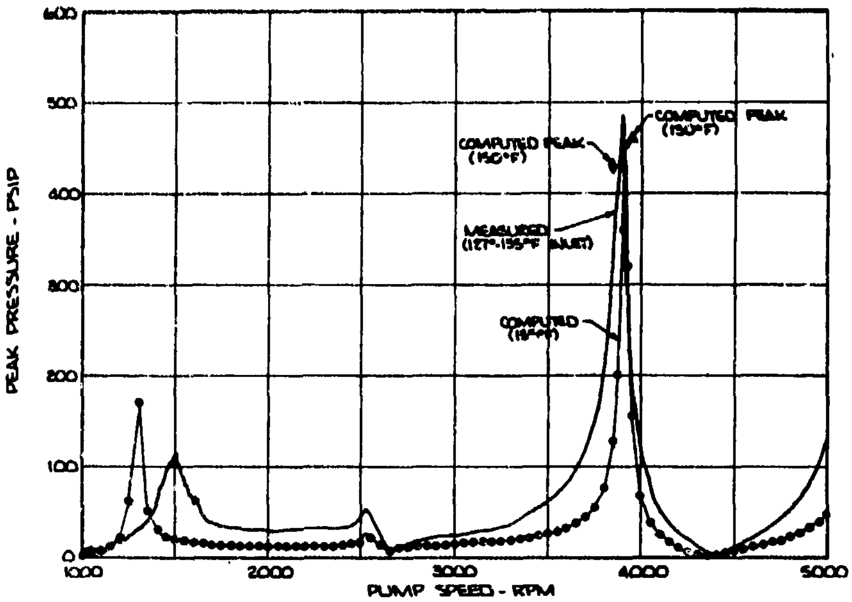
Note that computed peak pressures are shown for different temperatures. Temperature sensitivity analysis is discussed in Appendix B of Reference (3).

(2) Short Line Test Circuit Verification - An F-15 pump was installed in the test circuit shown in Figure 149. Piezoelectric type transducers were used to record pressure amplitude. The pressure measurements P1 through P5 were taken at 10 inch intervals along the line beginning at the upstream end of the 108 inch test section. Temperature was measured at the inlet of the test section.

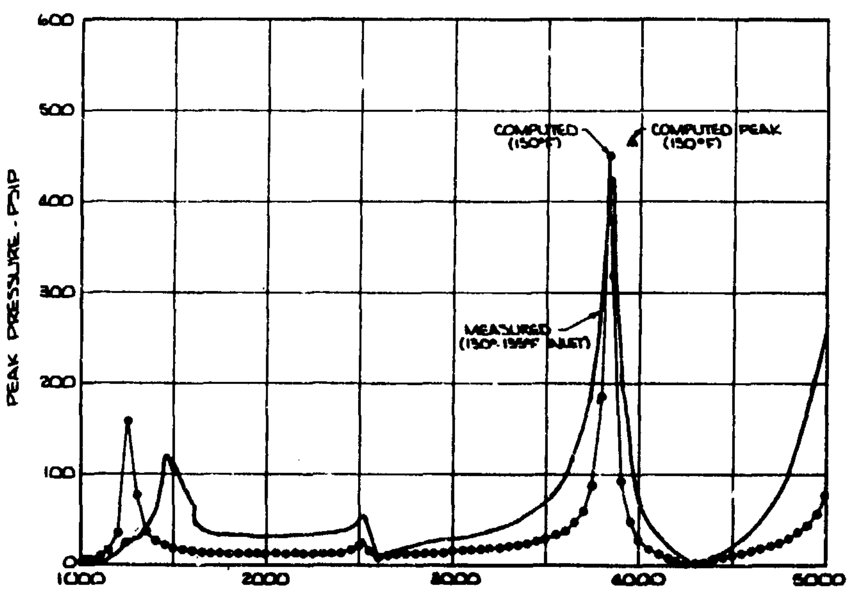
Data was recorded on analog tape as the pump speed was swept from 1000 to 5000 RPM. Data was played back from the analog tape through a signature ratio adapter to make X-Y plots of Fundamental Amplitude versus RPM.

The circuit was modeled for use with the revised HSFR program. Peak pressure amplitude values were computed at junctions in the circuit corresponding to the locations for which pressure measurements were made. The computed values were plotted on the recorded data for the fundamental frequency. The results are shown in Figure 149.

Correlation between predicted and measured values for this circuit is considered quite good, especially for the resonant frequency near 3900 RPM.



PRESSURE FREQUENCY RESPONSE
F-15 PUMP VERIFICATION TEST - 64-83-P1
MIL-H-5606B 3000 PSI 130°F 0.0 GPM



PRESSURE FREQUENCY RESPONSE
F-15 PUMP VERIFICATION TEST - 64-81-P1
MIL-H-5606B 3000 PSI 130°F 8.0 GPM

FIGURE 148 F-15 PUMP VERIFICATION TEST CORRELATION

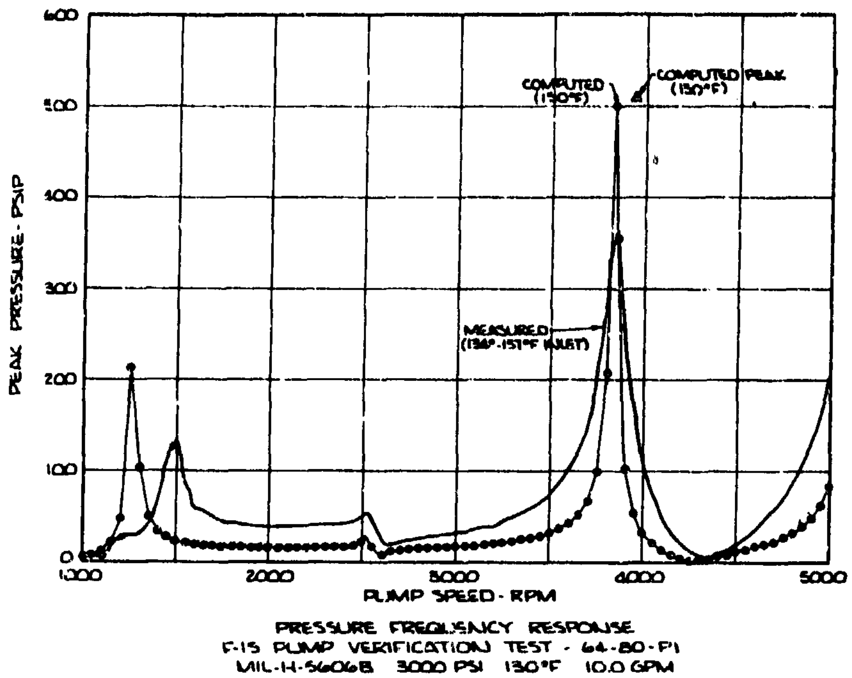
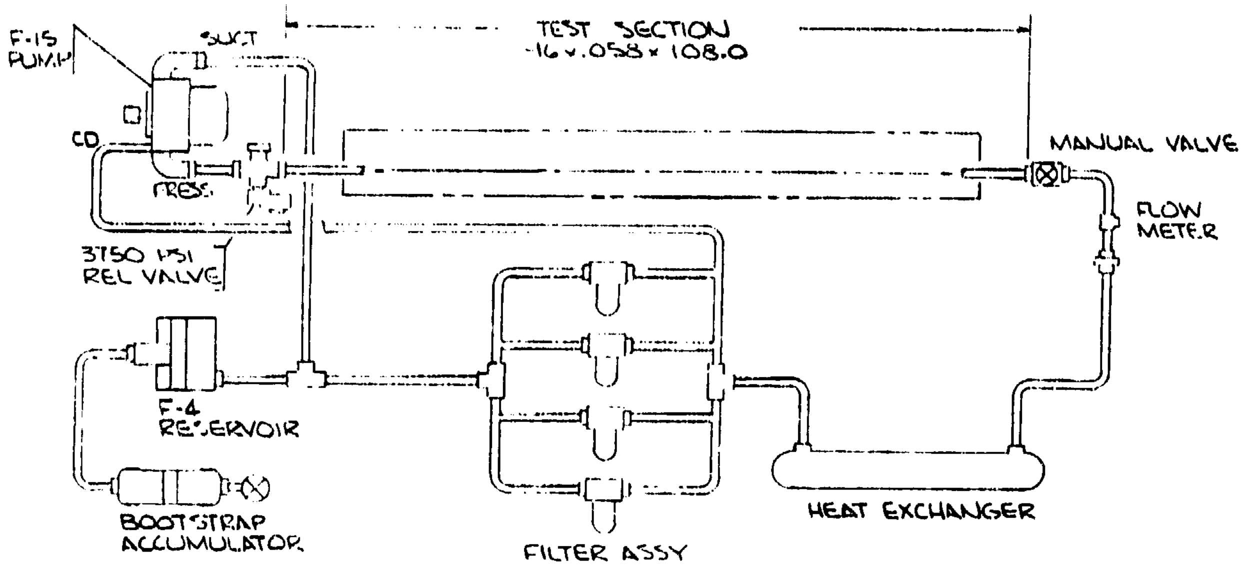
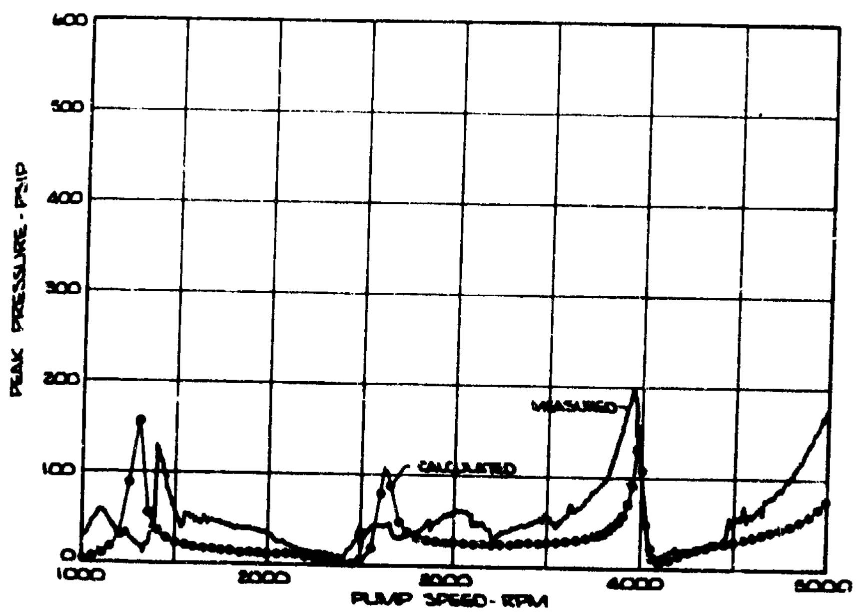


FIGURE 148 (CONCLUDED)

The bandwidth of the predicted resonant peak is narrower than the measured data indicates. However, a plot of the standing pressure wave, constructed from either the measured or predicted peak values, would yield a good picture of the pressure pulsation "signature" in the section. Interpretation of the data should be subject to the sensitivities discussed in Appendix B of Reference (3).

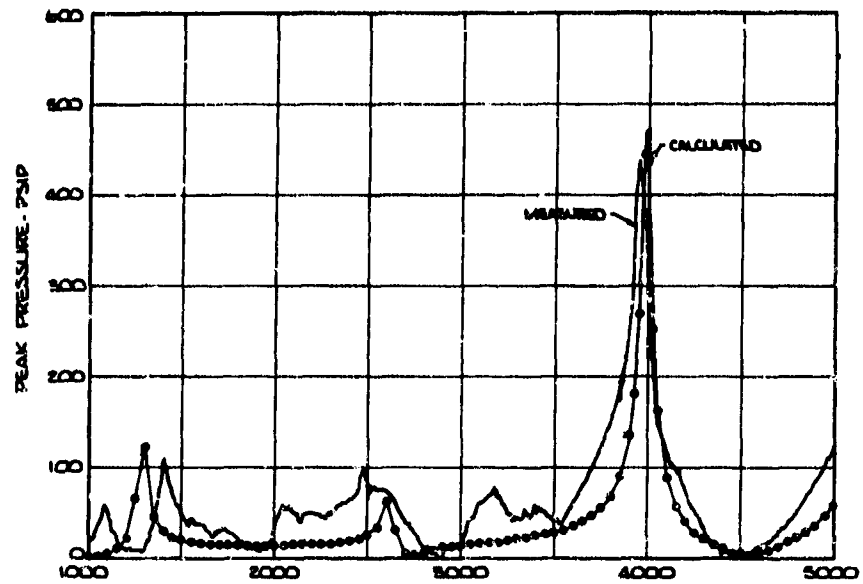


SHORT LINE TEST SETUP - RUN 109-6

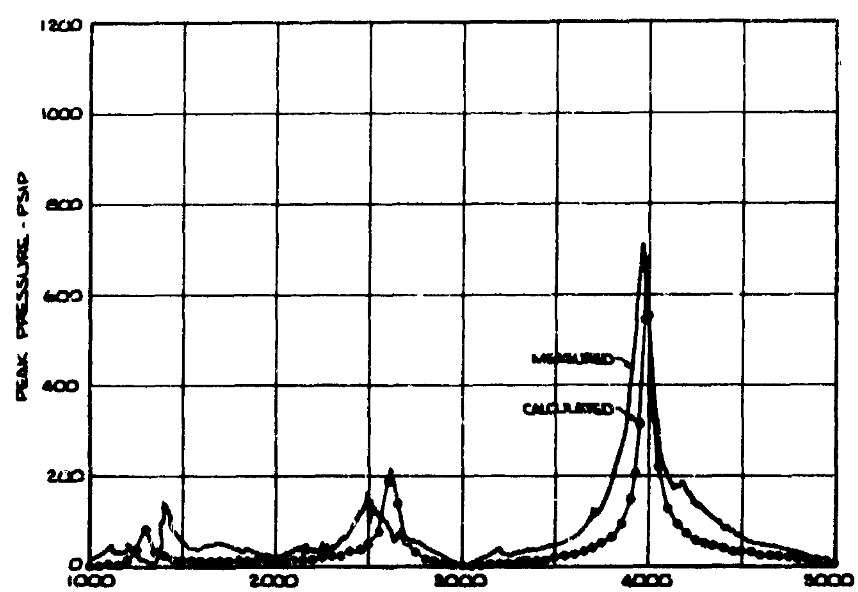


PRESSURE FREQUENCY RESPONSE
SHORT LINE TEST - 109-6-P1
MIL-H-5606B 3000 PSI 90°F ZERO FLOW

FIGURE 149 SHORT LINE TEST CIRCUIT CORRELATION

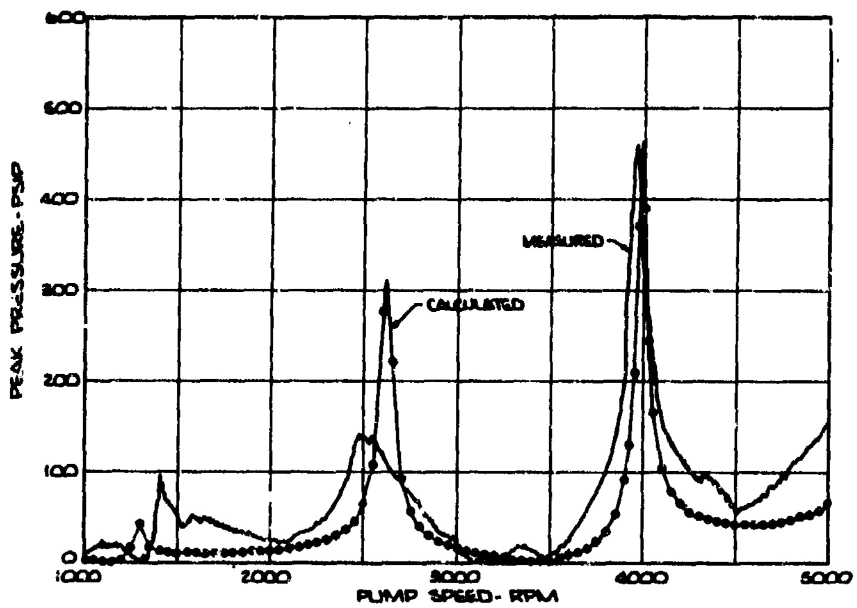


PRESSURE FREQUENCY RESPONSE
SHORT LINE TEST - 109-6-P2
MIL-H-5606B 3000 PSI 90°F ZERO FLOW

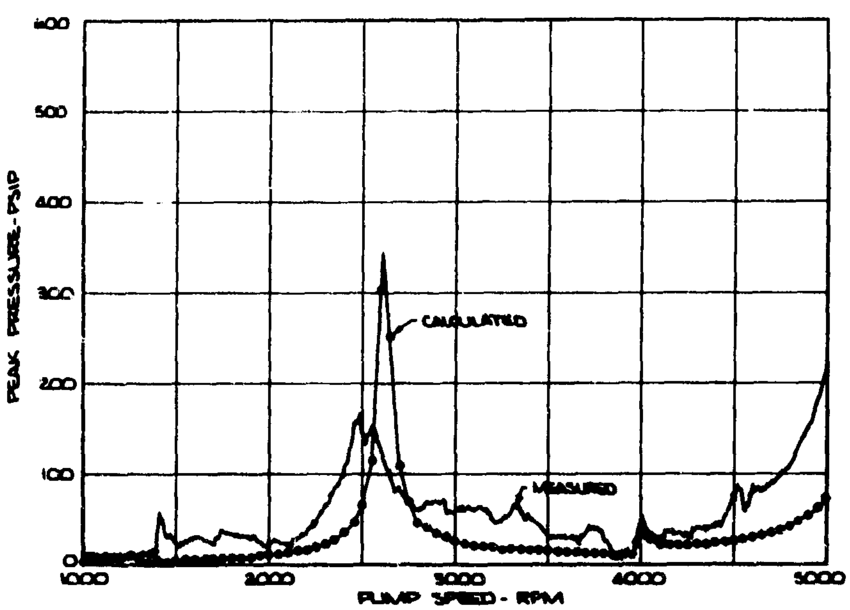


PRESSURE FREQUENCY RESPONSE
SHORT LINE TEST - 109-6-P8
MIL-H-5606B 3000 PSI 90°F ZERO FLOW

FIGURE 149 (CONTINUED)



PRESSURE FREQUENCY RESPONSE
SHORT LINE TEST - 109-6-P4
MIL-H-5606B 3000 PSI 90°F ZERO FLOW



PRESSURE FREQUENCY RESPONSE
SHORT LINE TEST - 109-6-P5
MIL-H-5606B 3000 PSI 90°F ZERO FLOW

FIGURE 149 (CONCLUDED)

3. ATTENUATION TECHNIQUES

a. Introduction

The purpose of this portion is to compile information regarding various techniques for the control of pressure pulsations in aircraft hydraulic systems. Information provided should serve as a guide in solving pulsation problems. Pulsation control techniques enumerated herein are drawn from the experience of MCAIR during the design, test, and manufacture of military jet fighters, and during the development, verification, and application of the Hydraulic System Frequency Response (HSFR) computer program.

b. Background

The ripple in the flow output of a piston-type hydraulic pump is transmitted throughout the supply side of the hydraulic system. The dynamic resistance (impedance) of the hydraulic fluid column creates pressure pulsations throughout the hydraulic supply system. The flow/pressure pulsations have a frequency of 750 hz for a nine-piston pump rotating at 5000 rpm.

When the frequency (speed) of the acoustic source (pump) is equal to a natural frequency of the hydraulic fluid column, a resonance condition occurs. Pressure pulsations at resonance may be very high; ± 1000 psi in a 3000 psi system. These resonant conditions are potentially very destructive to the system hardware. In extreme cases, the pressure vessel (lines or components) may reach fatigue failure limits in a few minutes. Hydraulic flow/pressure pulsations excite high frequency mechanical motion in lines and components. Excessive motion causes stresses which may result in failure of lines, components, and/or mounting hardware.

Pump discharge lines are generally the most vulnerable since they are in the area of maximum acoustic energy close to the pump. Dead-end lines to service ports and pressure transmitters are also highly vulnerable to pulsation induced damage.

Pressure pulsations also occur in the pump suction/return system. Normally, the suction system is not a problem because of the low pressure at which it operates. However, thin wall tubing, high installation stresses, and an adverse resonance condition could produce suction system failures.

c. Prediction and Measurement Of Hydraulic System Resonance Conditions

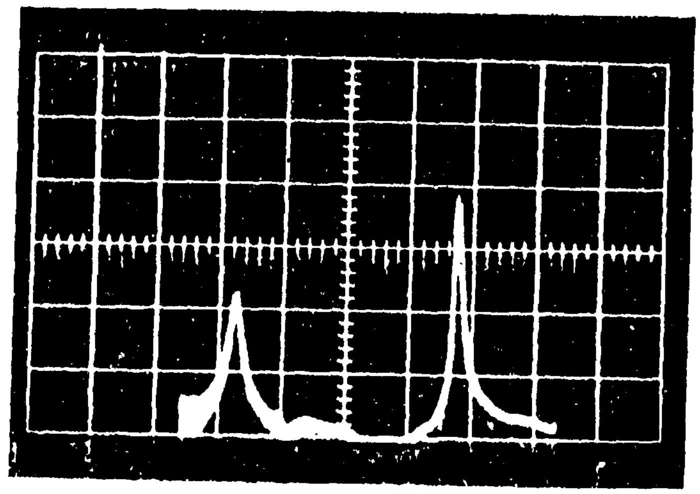
The Hydraulic System Frequency Response (HSFR) computer program was developed to predict the pump speeds at which hydraulic resonance occurs, and the amplitude of pressure pulsations at resonance. Use of the program is easy to learn with a couple weeks of on-the-job training and the guidance of the user's manual, (Reference 3). Interpretation of results for basic design purposes is also explained in the HSFR user's manual. Detailed understanding of the program theory and logic requires much longer and is documented in the HSFR technical manual (Reference 4). The program is relatively small and inexpensive to use.

Pump speeds at which system resonance occur are easily recognized from the computed output plots of pressure pulsations vs pump speed. Program output is formated the same as system pressure recorded on a spectrum analyzer during a sweep through the operating range of the pump (Reference 3).

If pressure pulsations at different locations in the pump discharge line are plotted for a given resonant speed, a standing pressure wave is the result. The standing wave is the result of superimposed incident and reflected acoustic waves in the hydraulic line. It is a key aspect to the accurate prediction and measurement of system pressure pulsations. The location of a pressure transducer only at the pump outlet usually does not give a true picture of maximum pressure pulsations in the system.

MCAIR developed and routinely uses a roving piezo-electric transducer to map standing waves in the system. It is clamped on the hydraulic line in enough accessible locations to define the standing wave and maximum pulsations at each resonant speed. Strain gage measurement of internal line pressures is a workable technique in the laboratory where calibration and zeroing before and after line installation can be controlled. However, it is difficult to obtain useful pressure data on an aircraft with this technique during ground and flight tests. Roving transducers can be used on the aircraft during ground tests to provide good pressure pulsation data.

Typical computed and measured pulsations are compared in Figures 150 and 151 from Reference 3. The HSFR program will automatically plot standing waves at the discretion of the user.



Spectrum Analyzer
Output - Run 28
Y-Axis 56 PSI/CM
X-Axis 500 RPM/CM
0-5000 RPM

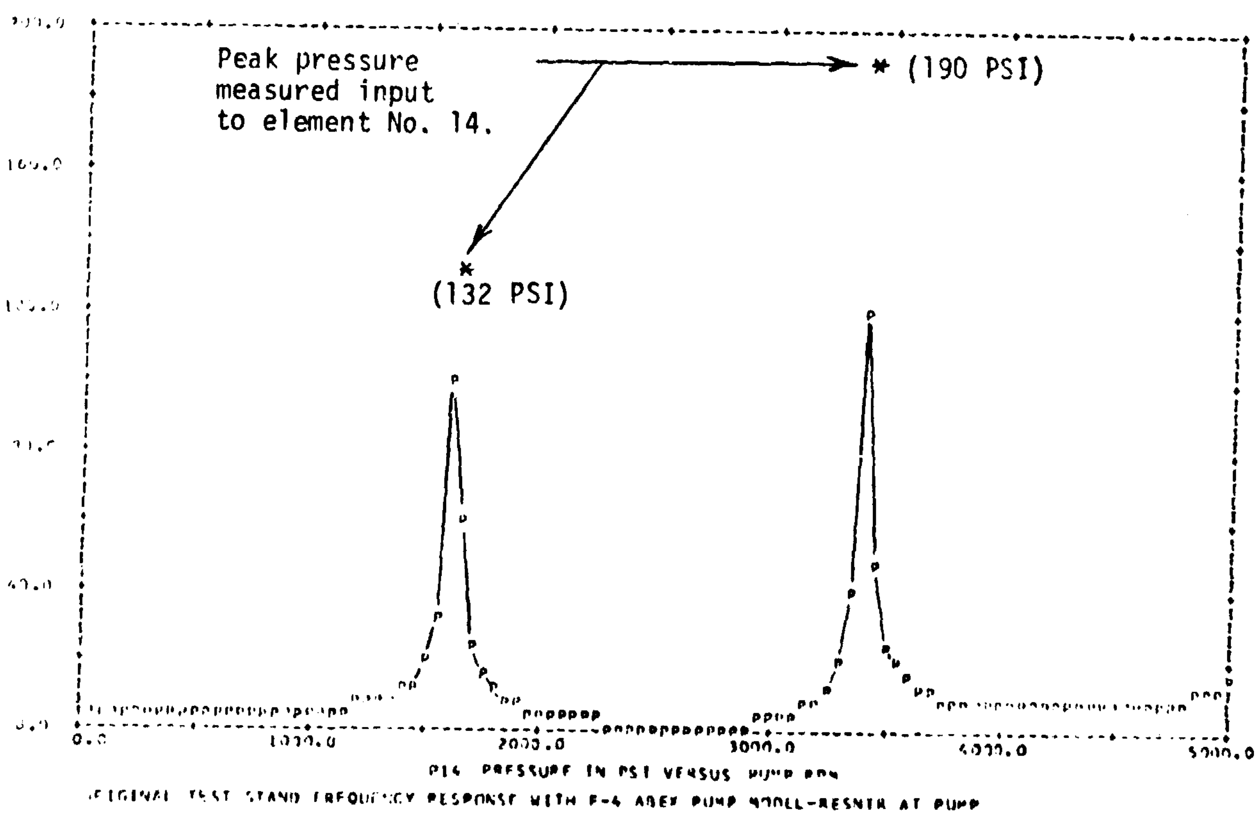


FIGURE 150 Computed and Measured Fundamental Frequency Response,
Test Stand F-4 Abex Pump

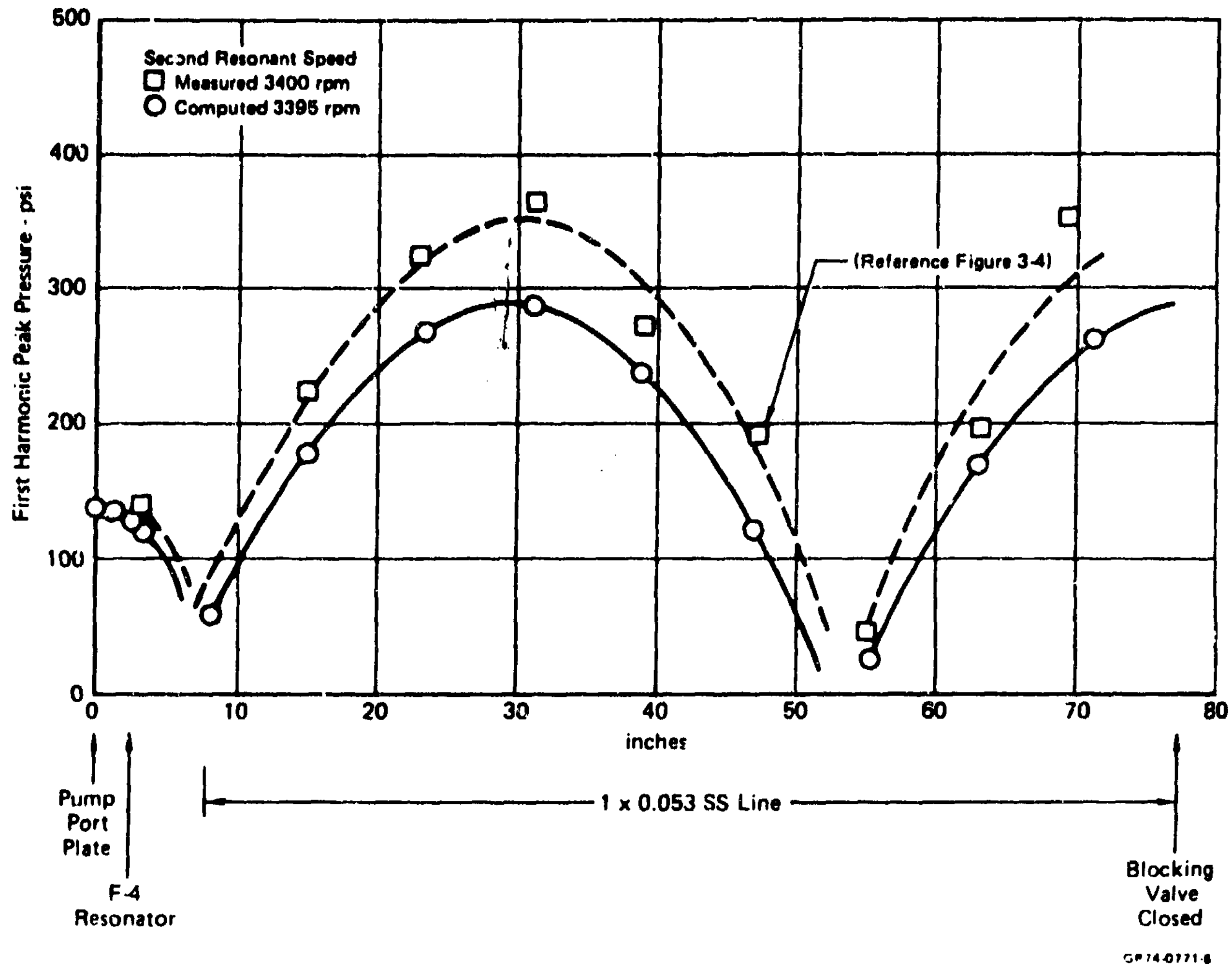


FIGURE 151 STANDING PRESSURE HALF WAVE, TEST STAND F-4 ABEX PUMP

The HSFR program is excellent at predicting the pump speeds at which hydraulic resonances occur. Amplitude prediction is usually pessimistic. Fortunately, this does not seriously hamper the usefulness of the program. System changes may be identified which will relocate resonant frequencies away from continuous operating pump speeds. Simple, short (<10 ft.) central systems with only two or three resonant speeds in the operating range, allow "room" for safe relocation of resonant speeds if there are only two or three continuous operating speeds. Fluid temperature and temperature changes are important. An 80°F increase in the temperature of MIL-H-5606 fluid will reduce system resonant speeds by 10% (3600 rpm from 4000 rpm).

d. Acceptable Levels Of Pressure Pulsations

The coupling of hydraulic to mechanical resonances is a complex phenomenon, and is a function of line size, configuration (routing), and installation constraints. Total stresses in lines are a combination of hoop stress from internal pressure and bending stress due to installation and induced vibration. Large pump discharge and suction lines (1 inch OD) are particularly vulnerable to high installation stress. Total combined line stress must be low enough to provide infinite fatigue life. Self-aligning fittings can virtually eliminate installation stresses.

An acceptable level of pressure pulsations in one system may not be acceptable in another system due to differences in mechanical response. MCAIR has determined via laboratory tests and F-15 experience that high frequency line motion induced by pressure pulsations cannot be controlled by normal line support techniques (clamps). Clamps must be designed to withstand the line vibration without wearing out the cushion and chaffing through the line. MCAIR laboratory tests and flight experience indicates the following relationships between central system pulsations and associated problems.

MAXIMUM PULSATIONS PEAK-PEAK	POTENTIAL PROBLEMS
> 600 psi	Rapid failure of pump discharge line due to pressure and vibrational stresses. Possible failures of mounting structure and internal functions of central components.
150 - 600 psi	Line clamp cushion wearout, line failure due to clamp chaffing, poor clamp life, frequent inspections required, discharge line check valve wearout.
< 150 psi	Trouble free, long life system.

The first category (> 600 psi) is a potential safety of flight situation. The second category is one of nuisance level problems which probably surface only after a considerable amount of operational flight experience. The best approach is to verify acceptable line stresses on the iron bird and the first flight aircraft to preclude failures of the first category. The computer program should be used early in the design stage to predict resonant speeds. Pump to filter line lengths or other simple plumbing changes may be identified to relocate resonances away from continuous operating speeds.

If stress levels are not acceptable, wideband attenuators must be considered, or hoses for mechanical decoupling.

Beyond the central system, i.e. away from the high acoustic energy of the pumps, gearboxes, and engines, the vibration environments are relatively benign. Good line support and adequate clearances between lines and lines and structure precludes significant vibration related problems.

e. Pulsation Control Techniques

Pressure pulsations in the central hydraulic system can be manipulated by several techniques which fall into two basic categories: (1) relocation of resonant frequencies, (2) wideband attenuation of pulsation amplitudes.

(1) Relocation of Resonant Frequencies

Techniques which relocate resonant frequencies in the central hydraulic system can range from simple changes in line lengths and sizes to the use of acoustic components such as Helmholz resonators. The primarily benefit of this type device is to relocate the system resonances away from continuous operating speed points; ideally above or below the operating speed range of the pump. A secondary benefit may be a reduction in pulsation amplitude. However, a reduction downstream of such a device may be accompanied by an increase in pulsations upstream of the device.

(a) Choke Line

The original installation of the F-15 PC systems produced a system hydraulic resonance at the cruise pump speed. Through an iron bird test effort and HSFR computer analysis, a solution to the problem was found. A 5/8 inch O.D. x 22 inch "choke" line was inserted just downstream of the pump outlet. This introduced a new reflection point in the system upstream of the pressure filter. Testing and HSFR analysis indicated that this change in a portion of the basic 1 inch size supply line moved the resonance point away from the cruise condition and reduced the pressure pulsation amplitudes (Fig. 152). F-15 production aircraft PC systems contain the "choke" line as a result of this effort.

(b) Volume

Emergency hydraulic pumps on the F-15 "Streak Eagle" aircraft were powered by constant speed D.C. motors. HSFR computer analysis indicated that steady state motor operating speed (3900-4000 rpm) was coincident with a system hydraulic resonance condition (Fig. 153). The HSFR program was used to analyze practical changes to the circuit. Installation of an available pressure filter and its optimum location in the system were determined by computer analysis. System hydraulic resonance was moved to 2937 rpm, comfortably below the minimum pump speed which resulted from variations in hydraulic load. Subsequent testing of the new configuration on the aircraft using roving transducers confirmed the computer predictions.

(c) Line Length

A frequency analysis of the F-18 hydraulic systems was first performed based on preliminary installation line data. The analysis showed a strong system resonance in the cruise to 100% pump speed range with fluid temperature variations expected during system warmup. The simplest and most cost effective approach was to change the line lengths between the pump and filter manifold to avoid resonances in the pump operating range (Fig. 154).

Optimum line lengths are of course subject to installation and line routing constraints. However, identification of optimum lengths early in the design state enhances the chance of incorporating it in the installation. The procedure was followed successfully during design of the emergency pump installation on F-18 flight test aircraft.

(d) Helmholz Resonator

The device basically induces a volume effect into the circuit which can significantly shift resonant frequencies. A 20 in^3 spherical volume resonator is attached to the pump outlet port on production F-4 hydraulic systems.

Location has a direct effect on the result. Figures 155 and 156 show maximum pulsations in a test circuit with an F-4 resonator mounted at different locations. Note the shift in resonant frequencies compared to the baseline test circuit without a resonator (Fig. 155). Figure 157 shows standing waves in one of the resonator test circuits. Note that pulsations are reduced downstream of the resonator but are still relatively high upstream.

(e) Accumulators

An accumulator in the central system produces a volume effect similar to filters and resonators. Significant shifts in resonant frequencies may be produced by adding or moving the location of an accumulator. Pulsation amplitudes may be reduced by specific accumulator design and use of the appropriate precharge. Standard MS accumulators generally are not effective.

(f) Hoses

Hoses are often mentioned as the solution to central system pulsation problems. While they do introduce "softness" to the circuit, pulsation amplitudes are not necessarily reduced by substituting hoses for central system hard lines. Pulsation amplitudes in the F-15 iron bird systems were about the same with hoses as they were with hard lines.

Hoses may provide mechanical decoupling which can eliminate clamp/line wearout problems. However, one must accept the penalties of limited hose life, added weight and cost, and larger diameters.

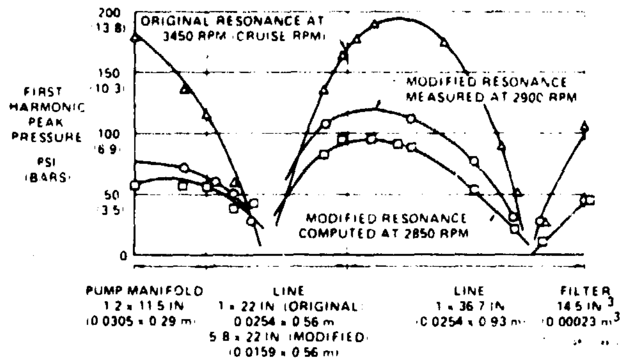


FIGURE 152 REDUCTION IN AMPLITUDE AND SHIFT OF RESONANT FREQUENCY ACHIEVED BY A CHANGE IN LINE SIZE

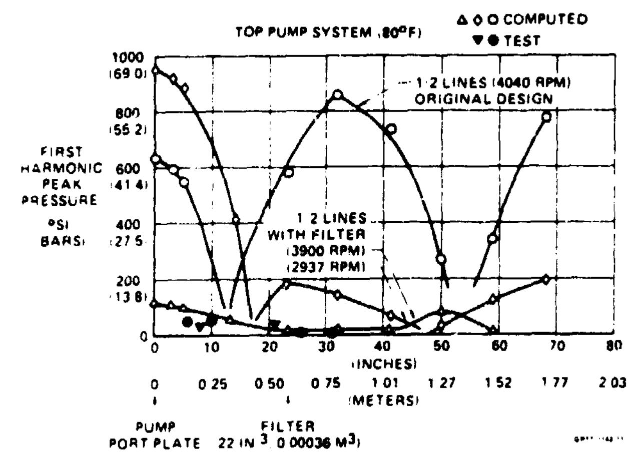


FIGURE 153 RELOCATION OF RESONANT FREQUENCY WITH ADDED FILTER, F-15 "STREAK EAGLE"

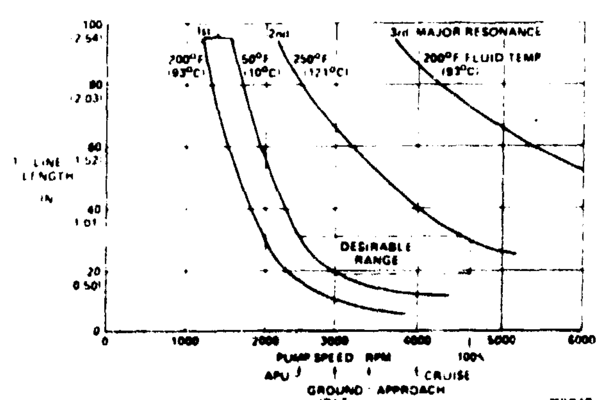


FIGURE 154 CONTROL OF CENTRAL SYSTEM MAJOR RESONANT FREQUENCY LOCATIONS BY LINE LENGTH

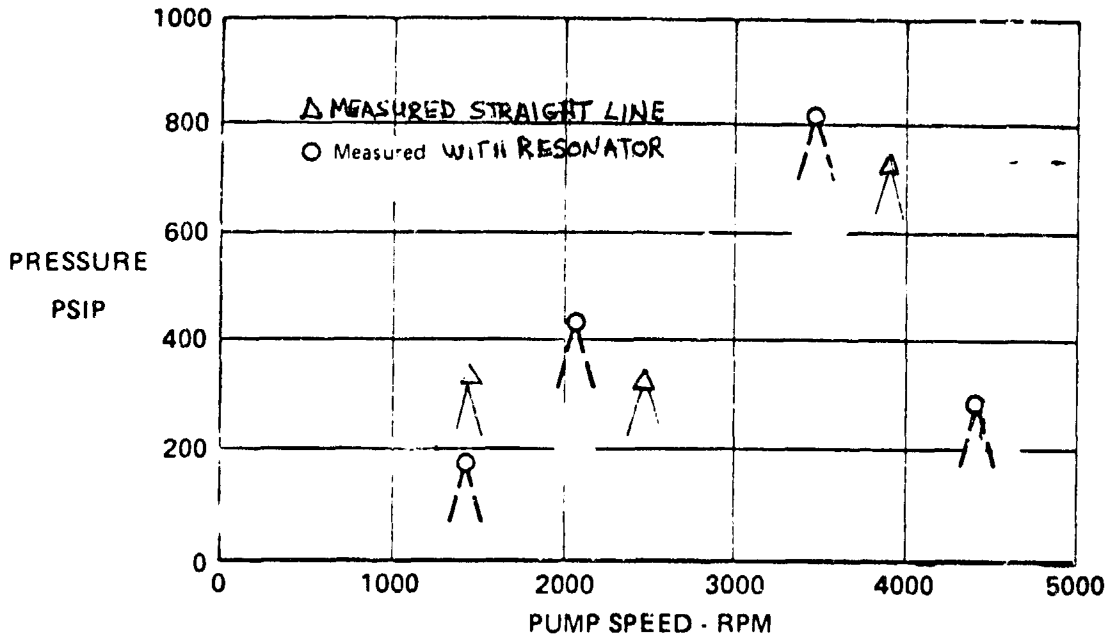


FIGURE 155 F-4 RESONATOR
DOWNSTREAM POSITION
MAXIMUM FUNDAMENTAL PEAK RESPONSE IN TEST CIRCUIT
MIL-H-83282, 130°F

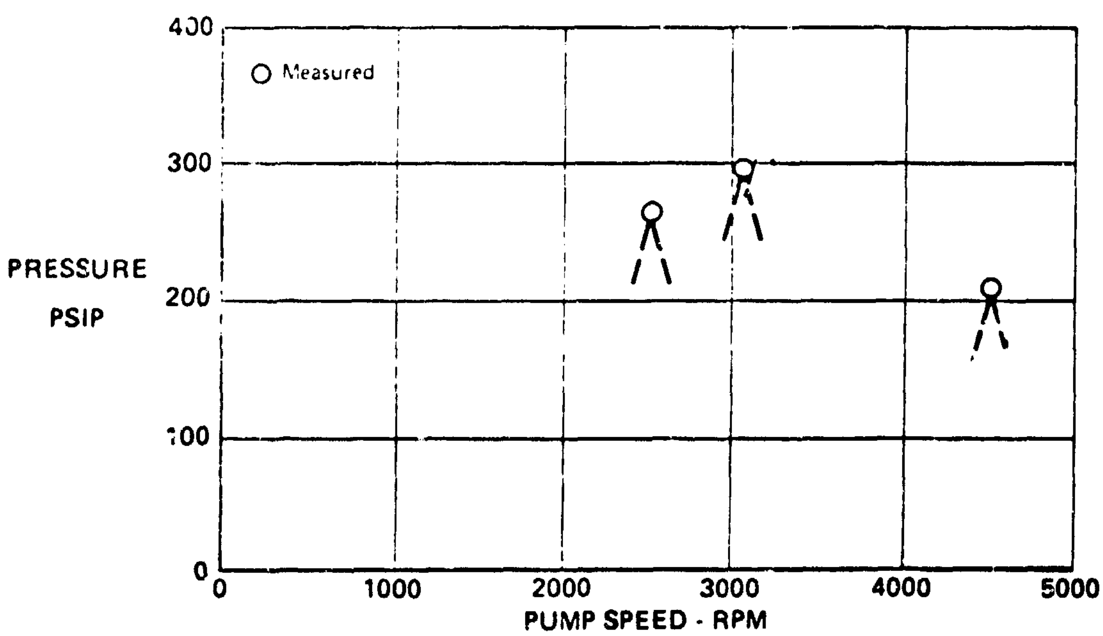


FIGURE 156 F-4 RESONATOR
UPSTREAM POSITION
MAXIMUM FUNDAMENTAL PEAK RESPONSE IN TEST CIRCUIT
MIL-H-83282, 130°F

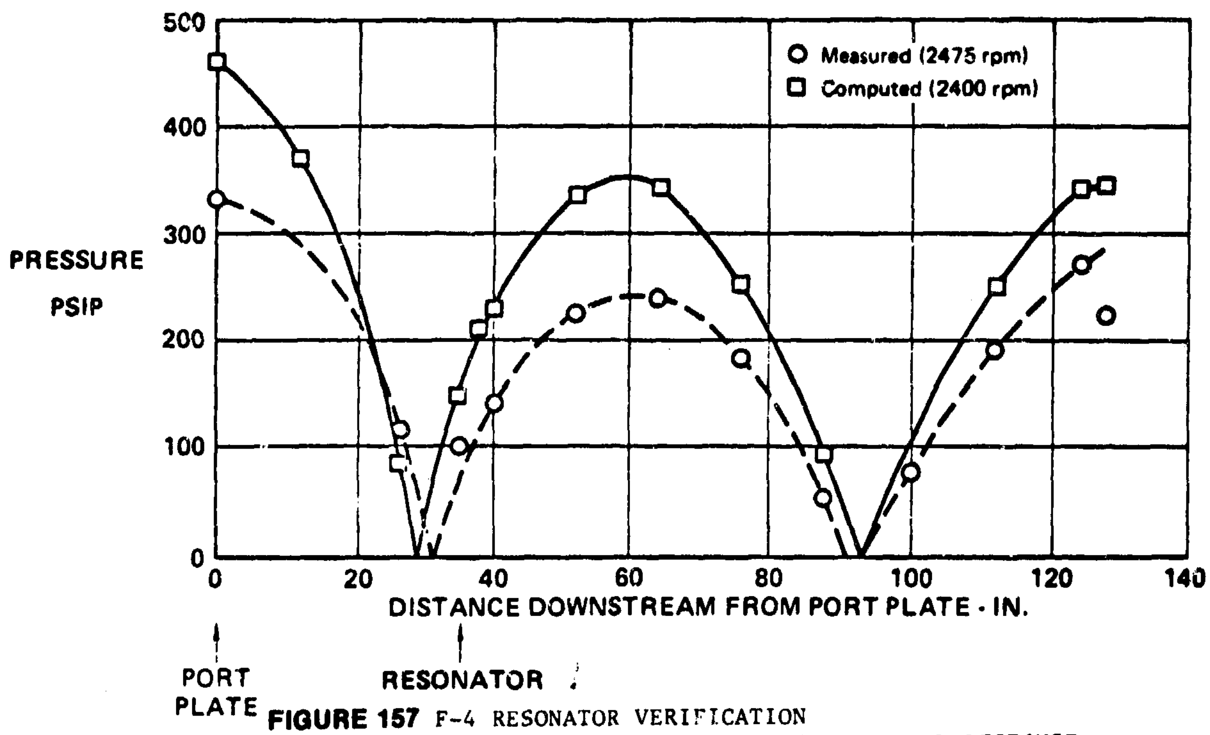


FIGURE 157 F-4 RESONATOR VERIFICATION
STANDING PRESSURE WAVE - FUNDAMENTAL RESPONSE
UPSTREAM POSITION
MIL-H-83282, 130°F

(2) Attenuation Of Pressure Pulsations

Techniques which significantly attenuate system pressure pulsations over the operating speed range of the pump are included in this category. Attenuation components added to the central system are the primary type in this category, however changes inside the pump are also included. Attenuators should be located as close to the source (pump) as possible for maximum protection of the downstream system. Pulsations at the pump outlet (upstream of attenuator) may or may not be reduced.

(a) Pulsco Attenuator

This patented device is shown schematically in Figure 158. It is designed, manufactured, and marketed commercially by the PULSCO Division, American Oil Filter Co., Louisville, Kentucky. MCAIR tested an ATP-1 model sized for a steady state ΔP of 100 psi at 10 gpm. Tests were performed on both a bench system and the F-15 iron bird left utility system to document performance and verify a computer model of the PULSCO unit.

Design cut-off frequency for the ATP-1 is 800 Hz (5333 rpm). Cut-off frequency is defined as the frequency above which attenuation of pressure pulsations is 90% or higher, i.e., the pressure pulsation level transmitted downstream of the unit is less than 10% of the input level. MCAIR tests showed significant attenuation above about 60% of cutoff frequency when the unit was installed at the pump manifold outlet. Figure 159 shows the effectiveness of the PULSCO unit when installed at the left hand utility pump on the F-15 iron bird. Pulsations were reduced from a maximum of 350 psi p-p (peak-peak) to about 60 psi p-p over the full range of pump speed (1000-4600 rpm). Pulsations upstream at the pump outlet were essentially unchanged. The natural frequency of the device (organ pipe frequency) was computed at approximately 2 times cutoff frequency, well above the using range.

A PULSCO unit sized for the F-15 application (100 psi ΔP at 50 gpm) is estimated to weigh about 6.2 lbs.

(b) Quincke Tube

MCAIR has developed a wide band attenuator based on the Herschel-Quincke tube. The concept is shown schematically in Figure 160 . Although sized for a single frequency (375 hz-2500 rpm), the device exhibits significant attenuation over a range of 2000-4500 rpm. This is adequate to cover the operating range of typical engine driven pumps used for primary hydraulic power.

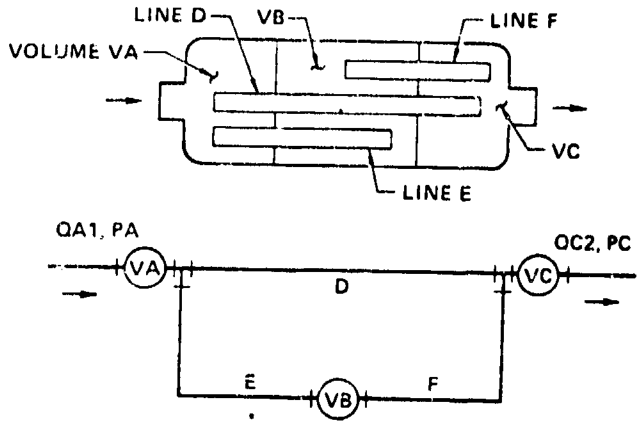


FIGURE 158 PULSCO HYDRAULIC ACOUSTIC FILTER COMPUTER MODEL FOR HSFR PROGRAM

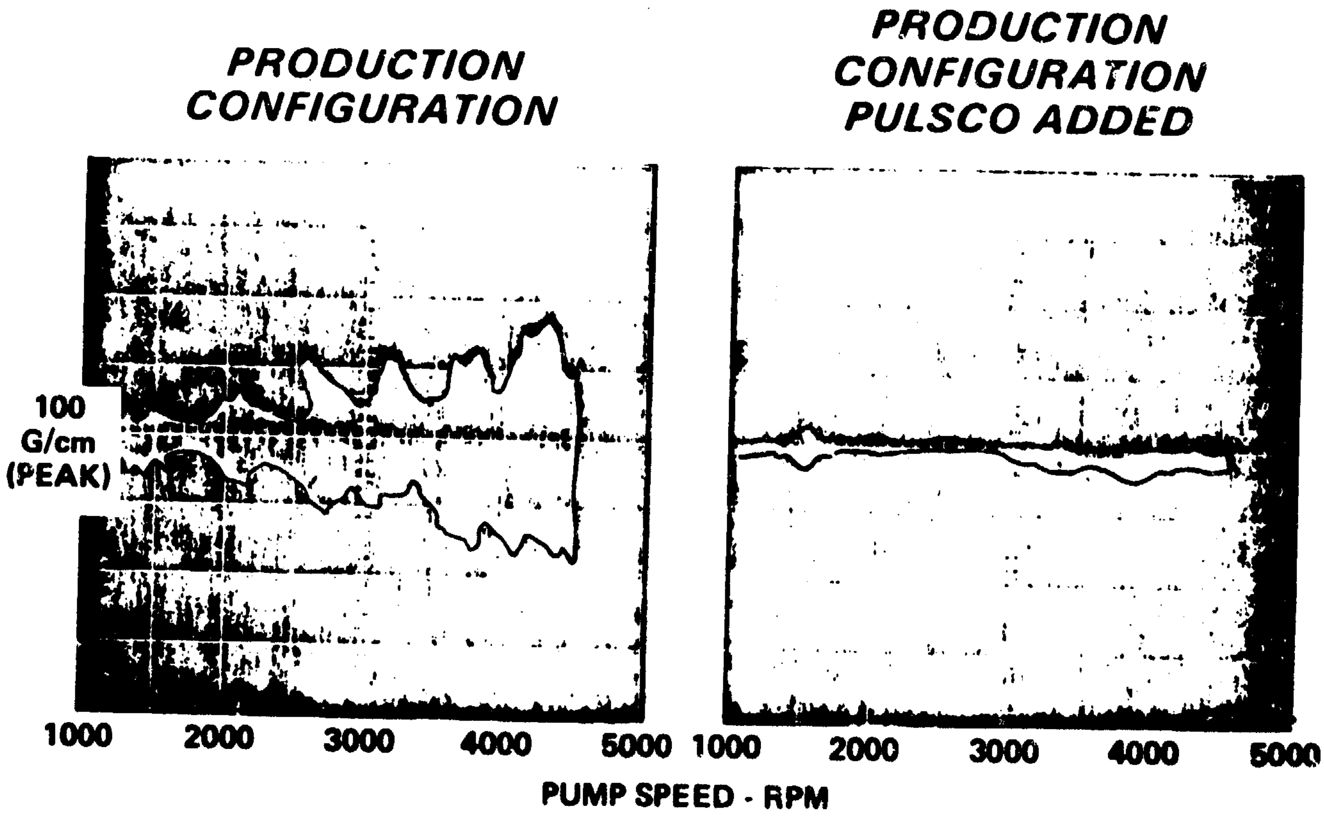


FIGURE 159 F-15 IRON BIRD - LEFT UTILITY SYSTEM TOTAL LINE ACCELERATION 18.5 IN. LOCATION - RIGHT PUMP OFF

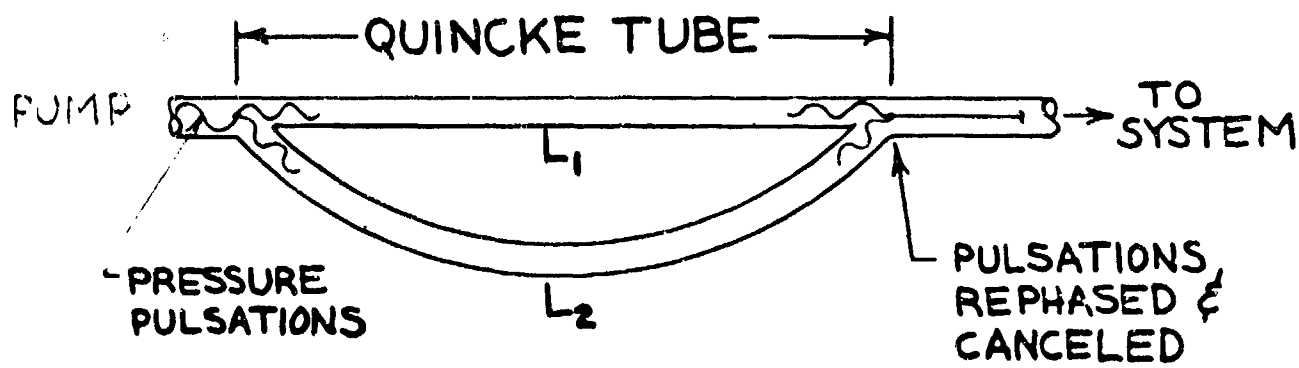


FIGURE 160 QUINCKE TUBE

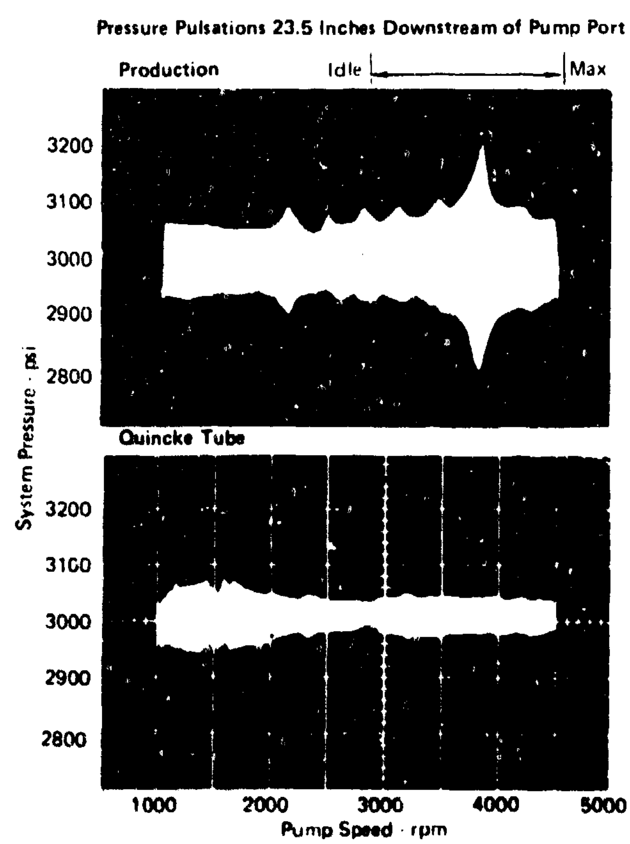


FIGURE 161 QUINCKE TUBE TEST RESULTS
F 15 Iron Bird L/H Utility System

Figure 161 shows pulsations in the left utility system of the F-15 iron bird with and without the MCAIR prototype Quincke tube. The unit was mounted directly to pump manifold discharge port. Attenuation performance was comparable to that of the PULSCO unit. Although adequate for the F-15 application, the effective range of the Quincke tube is not as wide as that obtained with the PULSCO unit. However, the MCAIR prototype Quincke tube weighs less than 1/3 as much as a PULSCO unit (1.7 vs 6.2 lbs) sized for the F-15 systems. The MCAIR unit was machined from titanium and has a smaller diameter (1.75 vs. 2.6 inches) and shorter length (10.8 vs. 13.0 inches) than the PULSCO unit.

The MCAIR prototype Quincke tube has performed consistently in several bench circuit configurations, one or both of the F-15 iron bird utility systems, and in a simulated F-18 system. Pressure pulsations upstream of the pump were never made any worse by the Quincke tube, and in some case were lower.

MCAIR developed and verified a computer model of the Quincke tube for HSFR program.

(c) Restrictors

Restrictors may be used to attenuate pulsations in a dead-ended line such as that terminated by a pressure transmitter. If a transmitter must be located at the end of a line, a small restrictor (.030 dia.) should be placed at the acoustic source, i.e. the "T" where the dead-end line begins. This protects the entire line and transmitter from significant acoustic energy. A secondary benefit is attenuation of system transient pressure spikes which may cause erroneous cockpit indications on a poorly damped pressure gage.

(d) Check Valves

A check valve may be used to acoustically isolate a dead-end ground service line which leads to the system pressure supply. Again, the check valve must be placed at the downstream end of the service line adjacent to the main line. This will isolate the service line from acoustic energy generated by the on-board system pump during normal aircraft operation.

(e) Pump Internal Effects

Several parameters internal to axial piston pumps effect its acoustic output. These are discussed below.

Inherent Flow Ripple

The summing of sinusoidal flow outputs from pump pistons produces an inherent pulsating or ripple character in the total output flow. The more pistons, the smaller the amplitude and the higher the frequency. The amplitude of the inherent flow ripple varies with flow output in a variable delivery pump. The pulsating outlet flow is the basic acoustic energy source which results in system pressure pulsations.

Pump Timing

Timing in an axial piston refers to the rotational position of the drive shaft as the rotating cylinder slot begins to communicate with a fixed slot in the port valve plate. It is desirable to minimize the mismatch of pressures between the cylinder and fixed slot as communication occurs. Mismatch causes a transient flow surge which contributes significantly to acoustic energy.

Pre-compression of the oil in the cylinder between closure of the cylinder to the suction slot and its opening to the outlet slot reduces the pressure mismatch and hence the outlet acoustic source energy. In a variable displacement pump, the optimum timing is usually set-up to occur at the flow condition most often encountered. This is steady state leakage flow on an aircraft hydraulic system.

Timing changes which occur transiently during higher flow demands can increase system pressure pulsations.

Special Features

Any internal feature which increases the mismatch (transient flow) during slot communication will adversely affect system pressure pulsations. One example is a by-pass slot to derate the flow delivery of a pump.

4.0 EMPIRICAL PUMP MODEL

Improved accuracy in HSFR component models depends on the elimination of test circuit effects. An objective of this effort was to develop a test circuit with very low standing wave amplitudes. The benefit of such development is not certain. However, it was considered feasible to develop a standard test circuit which would permit data measurement suitable for an empirical frequency model of a test pump of unknown internal characteristics.

a. Study Results

Further development of the empirical pump model depended on development of a resonant free acoustic source for accurate dynamic flow measurement in a non-reflective test circuit.

Better dynamic flow measurement is essential to deduction of pump shunt impedance because measurable steady state pump parameters alone cannot be used as input data. Acoustic source characteristics are dynamic and depend on many internal pump parameters; e.g., precompression,port plate valving geometry, as well as on steady state flow and outlet pressure.

The approach considered is to test the pump in a circuit of known low impedance. Pressure pulsations would be measured at all pump speeds. The resonant free acoustic source (for which dynamic flow characteristics would be known) would be tested in the same circuit over the same range of frequencies.

The circuit impedance can be calculated given dynamic pressure and flow characteristics using the accustic source. The dynamic flow of the pump can then be calculated given measured dynamic pressure.

Since the pump/circuit is modeled as an acoustic source with parallel shunt and circuit impedance loads, the shunt impedance may then be calculated.

Development of a resonant free acoustic source offers other potential Uses:

- Provide a source of measurable dynamic flow
- Allow accurate component model verification with a harmonic free acoustic signal of variable frequency
- Allow endurance testing to a specific acoustic environment
- Allow accurate measurement of load circuit impedance
- Provide calibration reference for evaluation of dynamic flowmeters

b. Verification

Efforts directed toward development of the acoustic source were plagued with mechanical problems. A simple, oscillating piston device driven by a hydraulic turbine was designed and built. The device was sized to generate pulsating flows in the range of ± 5 CIS.

The turbine driven shaft was supported at either end by close coupled journal bearings. An eccentric diameter between the bearings was used to drive a piston rod with a partial bearing. This in turn, drove the piston through $\pm$ .010 inch displacement. Outlet pressure acting on the face of the piston loaded the piston and piston rod on the shaft. All bearing surfaces and the piston were pressure lubricated through internal passages.

Initial testing of the device showed promising results. The output pressure waveform was nearly sinusoidal and lagged piston position by the expected 180°. However, an unexpected failure occurred early in the checkout runs. The main journal bearings siezed the shaft. Failure analysis suggested that the bearings had overheated.

The journal bearings were redesigned incorporating a thicker wall to act as a better heat sink. Pressurizing grooves were also incorporated to provide better lubrication flow to the bearing surface. The unit was reassembled and testing was resumed.

The failure repeated almost immediately. Repeated efforts to replace and rework bearings did not improve the endurance of the unit. The result was that no useful data was obtained. Therefore, further efforts to develop the non-reflective test circuit (required for empirical pump model improvement) were abandoned.

5.0 STANDING WAVE PLOT

Program usefulness has been extended by adding the capability to plot computed peak pressure values versus distance along a section of a circuit. The standing pressure wave thus produced has proven very useful in evaluating the pattern of pressure pulsations in the circuit. Preprogrammed write options have also been incorporated in the PUMP subroutine so that certain calculated parameters may be written for detailed analysis of pump performance. The standing pressure wave plotting capability is described in Appendix B. Write options are discussed in Appendix B of Reference (3).

SECTION IV
SSFAN PROGRAM

1. BACKGROUND

The SSFAN computer program was developed to analyze steady state flows in a closed aircraft hydraulic system, i.e., where the fluid leaving the pump circulates through the system either via a reservoir, or by direct line return, or both. The solution method used in SSFAN is a matrix type and is on one dimensional steady state flow throughout the network of legs in the system. SSFAN may be used to analyze existing systems, used as a preliminary design tool to predict system flows and pressures, and used to update new system designs as they are established.

2. INTRODUCTION

SSFAN has been updated to extend its calculation capability and reduced in size by simplifying the overall organization of the program. The addition of the Quasi-transient model allows prediction of subsystem operating times using time step increments. A capability to input various fluid temperatures at components giving a temperature distribution throughout the system has been added to give better simulation. New models for a fixed pressure and a fixed flow point expand the system analysis capability by allowing the designer to isolate a subsystem for analysis. The floating branch point extends the capability to have the pressures calculated at any point in the system.

3. QUASI-TRANSIENT MODEL

The SSFAN program may be run in a quasi-transient mode. With an input of time interval, time step and some additional component data, the quasi-transient model may be used to predict subsystem operating times. A computer graph subroutine has been added to plot flows, pressures and actuator position versus time. The input data for the time interval is the initial and final times. The time step is optional. If a time step is not input, a default value of 100 steps is used, which will provide a full set of points for the graph subroutine.

Required input data for valves is of two types, 1) valve position versus time and 2) valve position versus position of another component. The actuator input data includes load vs stroke positions and the initial actuator position. For a dynamic accumulator, the precharge conditions and the initial starting conditions are input. Hydraulic motor torque vs rpm and efficiency are required as additional data for a quasi-transient run. If additional data is not input for the above components, the quasi-transient program will get its data from the standard input data which means that the component load remains constant throughout the run.

The calculation method is as follows:

1) Balance the system using initial conditions at t_o
2) Increment time one Δt (time step) and calculate new positions for accumulators and actuators using $Q_o \times \Delta t = \Delta VOL$
3) Balance the system again using the new positions.
4) Calculate the average pressures (accumulators), loads (actuators) and torques (motors) from t_o to t_1.
5) Balance the system again using the average parameters from step 4)
6) $t_1 = t_o + \Delta t$. Balance the system to initialize conditions at t_1, and begin the next time step.

Example input and output data for the quasi-transient mode are shown in Technical Report AFAPL-TR-76-43, Volume V dated April 1980.

4. FLOW RESTRICTOR MODEL

The new flow restrictor (regualtor) model sets a constant flow rate in a leg of a system. The model breaks the leg,adds two branch points, fixes the flow out of the branch point at the restrictor inlet and fixes the same flow back into the branch point at the restrictor outlet. Pressures are calculated at both branch points. An orifice sizer model was added as a take off of the flow restrictor model because the orifice sizing feature just required an additional subroutine to size the orifice with the pressures output from the flow restrictor model. This orifice sizing method is described in Technical Report AFAPL-TR-76-43, Volume VI, Appendix D dated January 1980.

5. PROGRAM IMPROVEMENTS

a. New Component Models

Other new component models include a fixed pressure point, fixed flow point, floating branch point and hydraulic motor. The fixed pressure point and fixed flow point models are sub types of the accumulator type 7 model. The fixed pressure point model may be used to breakout a subsystem from the main system and set constant pressures at the inlet and outlet of the subsystem. The fixed flow point model is used to simulate flows to or from other subsystems in the system under study.

b. Existing Component Models

The tube bend pressure loss calculation was changed from an equivalent length (viscosity dependent) calculation to an energy loss calculation. An energy loss calculation was added to the cross model to consider splitting and combining flows. A single data point model was added to the special type 10 model so that a procurement specification component with a single design point for flow versus pressure drop could be input. The actuator model was changed to a two pressure point model and the calculation method was also simplified.

c. Program Simplification

The overall original program size was reduced from approximately 3500 lines to 2100 lines. This was done primarily in the sorting and assembly portion of the program. Data was originally stored in a separate array for each element (component). Now the data is stored in two arrays, one for static elements which have fixed resistance coefficients and the other for dynamic elements which are called during the calculation and iteration portion of the program and have resistance constants which change with flow rate or flow direction. This simplified the assembly search for matching junction numbers. The number of static resistance subroutines was reduced from approximately twenty five (one for each element) to three. The quasi-transient model added approximately 300 lines to the program for a total of 2400 lines.

d. System Capability

The program array sizes can be changed at one point in the main program to accommodate any size system under study. All subroutines use dummy array sizes for the variables dimensioned in the main program. A leg may have an unlimited number of dynamic elements. Previously, the limit was five dynamic elements. The program now uses a sparse matrix solution technique where the locations containing data for the matrix are stored and the locations containing zeroes are not stored. This reduces the calculation considerably for large systems because most of the data is located around the diagonal of the matrix with zeros for the other locations in the square matrix.

e. Stability Problems

The effort to identify stability problems associated with the iterative balancing resulted in the technique of assuming a flow direction through a component and using the resistance coefficient for the entire iterative balancing calculation. A comparison is then made for the actual flow direction to the assumed flow direction. If the assumed flow direction was incorrect, an indicator is placed in the data array for the particular component and an indicator is passed to the calculation subroutine signaling another balance is required. Components that are checked in this manner are check valves, relief valves, one-way restrictors, pumps and actuators.

The technique described above appears to have solved some of the stability problems previously encountered in SSFAN. However, there still may be other problems. One occurrence that has not been fully investigated appears to be associated with a very small change in flow in a leg creating a large change in pressure at the ends of the leg.

6. SSFAN DOCUMENTATION

Consult References (7) and (8) for the user and technical descriptions of the Quasi-transient model, the flow restrictor model and the other program improvements.

SECTION V

HYTTHA PROGRAM

1. BACKGROUND

The HYTTHA program is used to simulate thermal effects in aircraft hydraulic systems. The program calculates flowrates, pressures, and temperatures throughout the system.

Use of the HYTTHA program requires detailed physical information about components in the hydraulic system. The physical information is needed to determine heat transfer to or from the components. The significance of the required information varies with the problem being analyzed. Approximations for the required information, or options in the program coding, to allow for approximations, can simplify use of HYTTHA. These two simplification approaches were investigated.

Using approximations for input data would require creating more generalized subroutines than those already existing. The general information about the thermal and physical component characteristics would be input by the user. Because of the general nature of these routines, they would be more unreliable.

However, the user has the option of including detailed physical information not in the component models. Length and width parameters can be estimated to simulate box type components. Heat transfer coefficients can be estimated from textbook approximations. Where heat transfer between the thermal environment and the hydraulics is negligible, these terms could be zeroed. Reasonable simulations can be obtained with minimal input data.

Consequently, program coding options are the most feasible approach to simplifying the HYTTHA program. Three features were added to the HYTTHA program, a default temperature section, an environment section and a modified reservoir component subroutine for defining thermal characteristics. Several program improvements including a variable RPM section for the pump and an energy loss coefficient for line bends were added to HYTTHA and are explained in the following sections.

2. PROGRAM IMPROVEMENTS

a. Default Temperature Section

This program modification is an efficient way of changing large amounts of input temperature data. Each individual line and component has four default temperatures (ambient, structural, wall and fluid). The user can make the program default the line or component to the corresponding temperature dependent upon the value of the default temperature indicators.

The default temperature section has the capability of setting all line and component temperatures to the respective default temperatures (ambient, structural, wall, and fluid). The program has a master indicator and each line and component has indicators to skip the section if desired. There is also an indicator within each element to skip any specified temperature the user prefers not to change.

b. Environment Section

The environment section allows the input of up to five different environments (the maximum number of environments can be changed to accommodate more than five). Each line and component can select one of the input environments. The program will change the element's initial ambient, structural, and wall temperature to the appropriate temperature in the specified environment.

An example of an environment is the temperature in the landing gear bay during a normal aircraft mission. Each line or component will choose the appropriate environment which will remain constant for the entire simulation unless changed by the particular component.

c. Constant Pressure Reservoir Model

The constant pressure reservoir subroutine (TRSVR61) has been modified to allow the user to input a fluid temperature curve. The user and technical manual write-ups are in Appendix R.

d. Pump Model - TPUMP51

Model steady state computation stability and a pump start-up capability were investigated.

The work on pump stability was focused on better initialization and improving the case drain section. Several methods were tried to start the iteration process with better flow and pressure values. Both initializing pressures and flows in the program and in the leg data have been tried. One method that has worked was to set the pump and reservoir node pressures in the leg block data similar to setting a constant pressure source. This helped to keep the values for successive iterations in the general area, but it did not damp out the oscillations.

Work in the case drain section has involved trying to desensitize the flow dependency of the model. Changing the weighting of old and new case pressures, updating the case pressure only every ten iterations, and setting the case pressure to 250 psi have all been tried. Large negative flows were obtained when the case pressure was updated every 10 iterations. When the case pressure was set to 250 psi, a solution was obtained having large case drain flows and negative case pressures. Setting the case pressure to 250 psi was effective but it had some difficulties.

A brief look was made at the possibility of a three node pump, but no major work was done with this approach.

Several changes have been made in the HYTTHA pump model steady state section. After attempting to stabilize the existing model, the steady state section was rewritten using the SSFAN pump model as a guide.

The major changes in the steady state section were to do all of the pump calculations in one section and to set the pump case pressure independently of the case leg pressure drop. As the case pressure increases, the case flow decreases due to internal leakage back to the pump inlet. But if the case pressure is used as the case leg pressure drop, the case drain flow will increase with case pressure. By first finding the case pressure and flow and then setting a separate case leg pressure drop, the proper case pressure - flow relationship can be obtained.

The pump model was modified to simulate pump start up characteristics. The user inputs a time history of pump speed with the pump component data. The subroutine will linearly interpret the data to obtain the pump speed for any time during start-up or shut down simulations with the HYTTHA program.

In the past, instabilities have occurred in the pump outlet pressure and flow at very high system flows. The instability was a result of the rapid decrease in pump outlet pressure at outlet flows greater than the rated pump flow. To eliminate the instability, the pump routine was revised to calculate steady state flows for the system based upon the relatively constant pump pressure for flows below rated flow. After a balance is obtained, the outlet flow is checked to see if it is greater than rated flow. If outlet flow is greater than rated flow, the steady state flows and pressures are recalculated using the pump outlet flow - pressure relationship for high pump flows.

The pump model changes are documented in the user and technical manual sections of the HYTTHA program. The revised sections are presented in Appendix S.

e. Calculation Subroutines - TCALC and TLEGCAL

The HYTTHA calculation subroutine (TCALC) has been revised to include a new compressed matrix method. The method reduces the amount of storage space required to solve for the steady state pressure values.

The pressure drop in a leg was dependent on the upstream fluid temperature. This section in TLEGCAL was changed to average the upstream and downstream temperatures. The average line temperature is now used to determine the line pressure drop.

The programming changes are documented in Appendix T.

f. LINE MODEL - TLINEA

The equivalent line method for determining steady state pressure losses due to bends in hydraulic lines is inaccurate at lower temperatures, high fluid viscosity regions. A more accurate method is computing an energy loss coefficient which is a function of fluid density, pressure drop and fluid velocity in a bend. The bend energy loss coefficient calculation used in HYTTHA is based on empirical test data. See Appendix U for the HYTTHA user and technical manual changes.

g. OUTPUT OPTIONS

The control data cards needed to use the default temperature and environment sections are explained in Appendix V. In addition steady state print options and y-axis scaling are explained in the appendix. The card column locations of the component default and environment indicators are shown in Table 22.

TABLE 22 COLUMN LOCATION OF COMPONENT DEFAULT

AND ENVIRONMENT INDICATORS

THE INDICATORS ARE ENTERED ON THE COMPONENT INTEGER DATA CARD

COMPONENT	TYPE NUMBER	DEFAULT TEMPERATURE INDICATOR* COLUMN	ENVIRONMENT NUMBER COLUMN
FRICTIONLESS BRANCH	11	40	45
TWO-WAY CONTROL VALVE	21	35	40
FOUR-WAY CONTROL VALVE	22	45	50
CHECK VALVE	31	30	35
RESTRICTOR	41	30	35
PUMP	51	60	65
CONSTANT PRESSURE RESERVOIR	61	40	45
BOOTSTRAP RESERVOIR	62	45	50
HEAT EXCHANGER	69	30	35
ACCUMULATOR	71	30	35
FILTER	81	30	35
VALVE CONTROLLED ACTUATOR	101	35	40
UTILITY ACTUATOR	102	30	35

* 0 - Uses temperature values on component cards

1 - Uses default values from Card 2a

3. VERIFICATION TESTS

The test objective was to measure the thermal response of the hydraulic fluid and components in an operating hydraulic system. Steady state and transient thermal data was recorded for seven test runs.

a. Test Setup and Instrumentation

A photograph of the thermal test bench is shown in Figure 162. A system schematic is presented in Figure 163 and Table 23 gives the appropriate line and component lengths. The system was powered by a F-15 hydraulic pump. The primary heat load came from the load valve. A heat exchanger cooled the hydraulic fluid and provided temperature stabilization for the steady state tests. The test stand was equipped with a 3750 psig relief valve and four A/C filters with 10μ nominal and 25μ absolute filtration. An accumulator provided a reference pressure for the F-4 bootstrap reservoir.

The pump and load valve installations are detailed in Figures 164 and 165. the heat exchanger installation photograph is shown in Figure 166

Except for the heat exchanger, the test setup was enclosed in a styrofoam box. Six thermocouples placed inside the box monitored the box air temperature. A Thermatron temperature controller was used to maintain the system environment temperature. Cooling air was injected by the F-15 pump and exhausted at the heat exchanger end of the box.

The instrumentation locations are shown in Figure 163 The parameters recorded during the testing are presented in Table 24.

To accurately measure the hydraulic systems steady state and transient thermal response, Type T thermocouples were placed in the fluid stream and on the component surface. The inline thermocouples were located at approximately the tube centerline. Each probe contained two copper constantan wires supported by a Conax fitting (Figure 167). The surface thermocouples were tack welded to the component.

Since each instrument position contained two thermocouples, one thermocouple was displayed on a stripchart recorder for a real time check on the data.

The other thermocouple went to an analog tape recorder through a 150°F reference box. A calibration was run before each test to assure that the thermocouples did not drift from their reference values.

The copper constantan 28 gauge thermocouple wire gave good temperature responses and sensitivity.

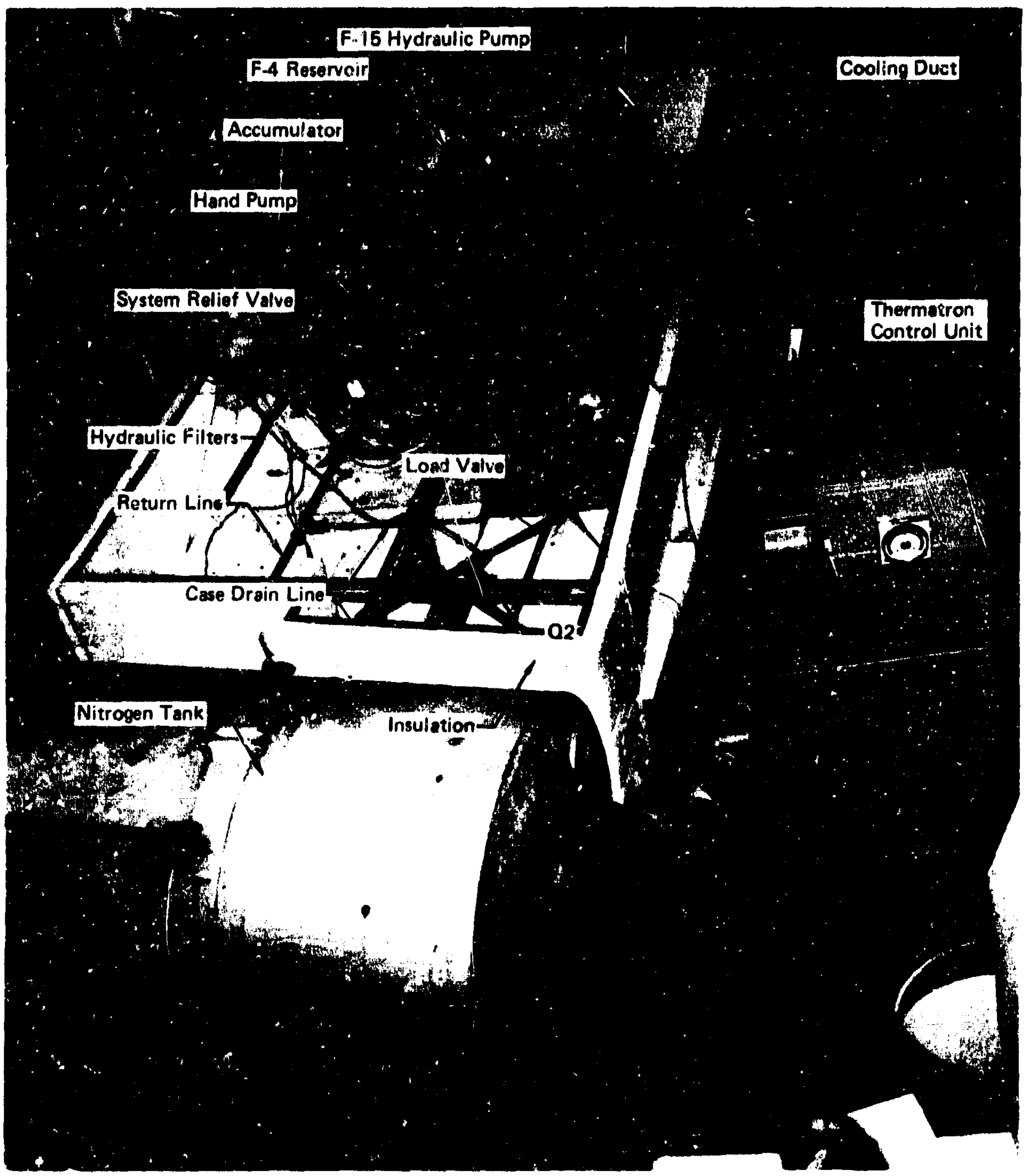
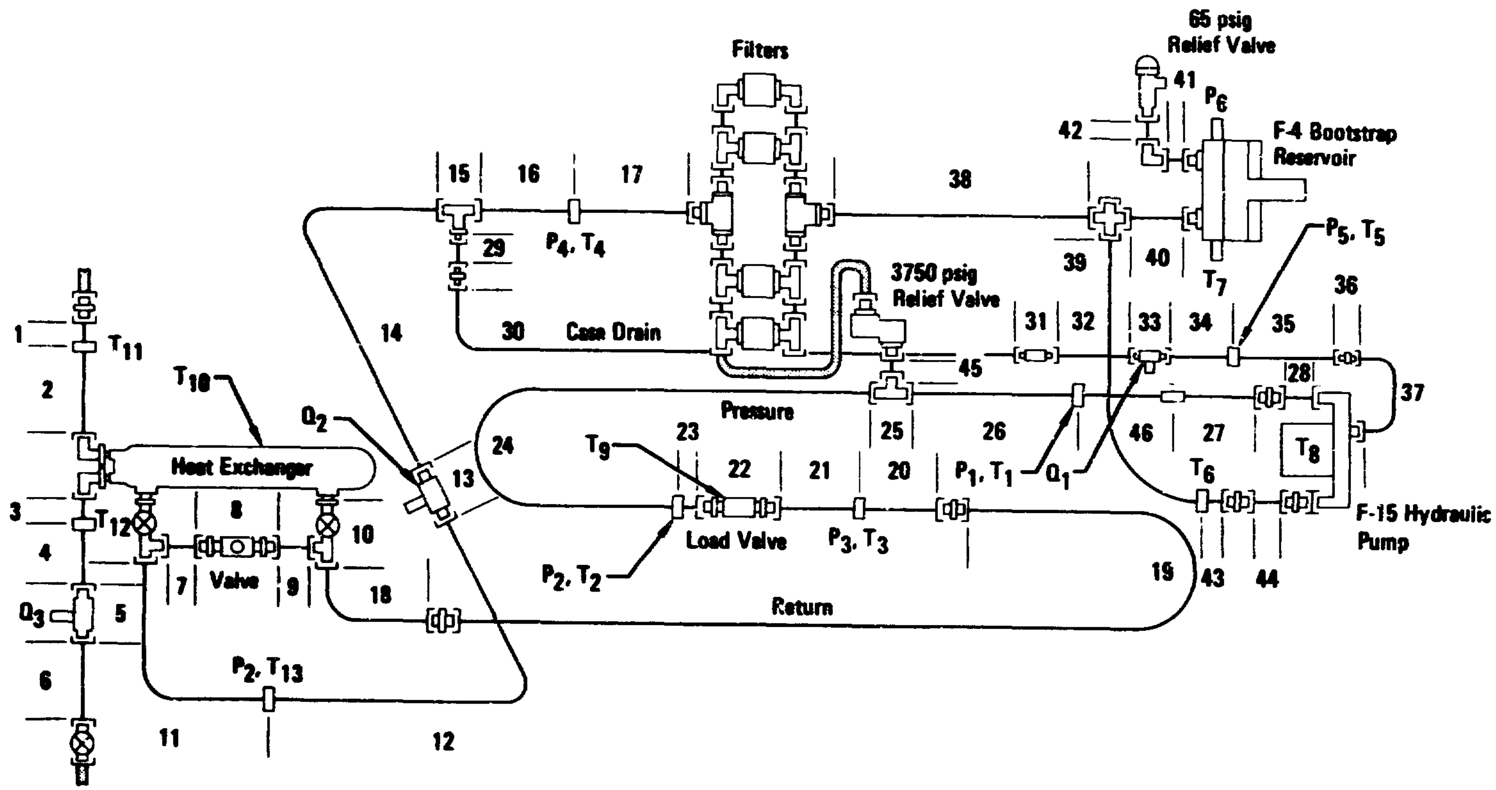


FIGURE 102 HYTTHA TEST BENCH



Notes: 1. All lines are 304 stainless steel.
2. Line thermocouples are mounted in fluid stream.
3. Heat exchanger to be excluded from cold box setup.
4. Filter lines are 0.75 in. O.D. x 0.042 in. wall thickness.

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FIGURE 163 HYTTHA STEADY STATE AND TRANSIENT TEST BENCH LINE LENGTHS AND INSTRUMENT LOCATION

TABLE 23 HYTTHA LINE AND COMPONENT LENGTHS

ALL LINES ARE 1" OD X .058" WALL THICKNESS UNLESS OTHERWISE NOTED

NUMBER	LENGTH (INCHES) ±.0625"	NUMBER	LENGTH (INCHES) ±.0625"
1	8.625	21	15.125
2	15.25	22 LOAD VALVE	5.25
3	15.125	23	1.625
4	16.00	24	144.125
5 FLOW METER	3.25	25 TEE	3.50
6	7.875	26	53.875
7	11.00	27	2.25
8 VALVE	7.125	28	4.875
9	11.25	29	3.375
10	6.875	30*	75.00
11	25.50	31 CHECK VALVE	2.875
12	43.625	32*	1.625
13 FLOW METER	3.375	33 FLOWMETER	2.875
14	6.625	34*	15.125
15 TEE	3.375	35*	20.875
16	15.25	36 UNION	1.25
17	15.125	37*	5.875
18	15.625	38	40.875
19	142.50	39	25.125
20	21.875	40	3.125
		41	2.75
		42	4.25
		43	3.50
		44	6.75
		45	5.50
		46	13.375

* 0.375" O.D. X .022" WALL THICKNESS

TABLE 24 HYTTHA RECORDED TEST PARAMETERS

PARAMETER	DESCRIPTION
P1	PUMP OUTLET PRESSURE
P2	PRESSURE UPSTREAM OF LOAD VALVE
P3	PRESSURE DOWNSTREAM OF LOAD VALVE
P4	RETURN PRESSURE
P5	CASE DRAIN PRESSURE
P6	RESERVOIR PRESSURE
Q1	CASE DRAIN FLOW
Q2	RETURN FLOW
Q3	WATER FLOW
T1	PUMP OUTLET FLUID TEMPERATURE
T2	FLUID TEMPERATURE UPSTREAM OF LOAD VALVE
T3	FLUID TEMPERATURE DOWNSTREAM OF LOAD VALVE
T4	FLUID TEMPERATURE UPSTREAM OF FILTERS
T5	CASE DRAIN FLUID TEMPERATURE
T6	SUCTION FLUID TEMPERATURE
T7	RESERVOIR FLUID TEMPERATURE
T8	PUMP CASE TEMPERATURE
T9	LOAD VALVE TEMPERATURE
T10	HEAT EXCHANGER SURFACE TEMPERATURE
T11	HEAT EXCHANGER OUTLET WATER TEMPERATURE
T12	HEAT EXCHANGER INLET WATER TEMPERATURE
T13	FLUID TEMPERATURE DOWNSTREAM OF HEAT EXCHANGER
DS	DRIVE SPEED
DT	DRIVE TORQUE

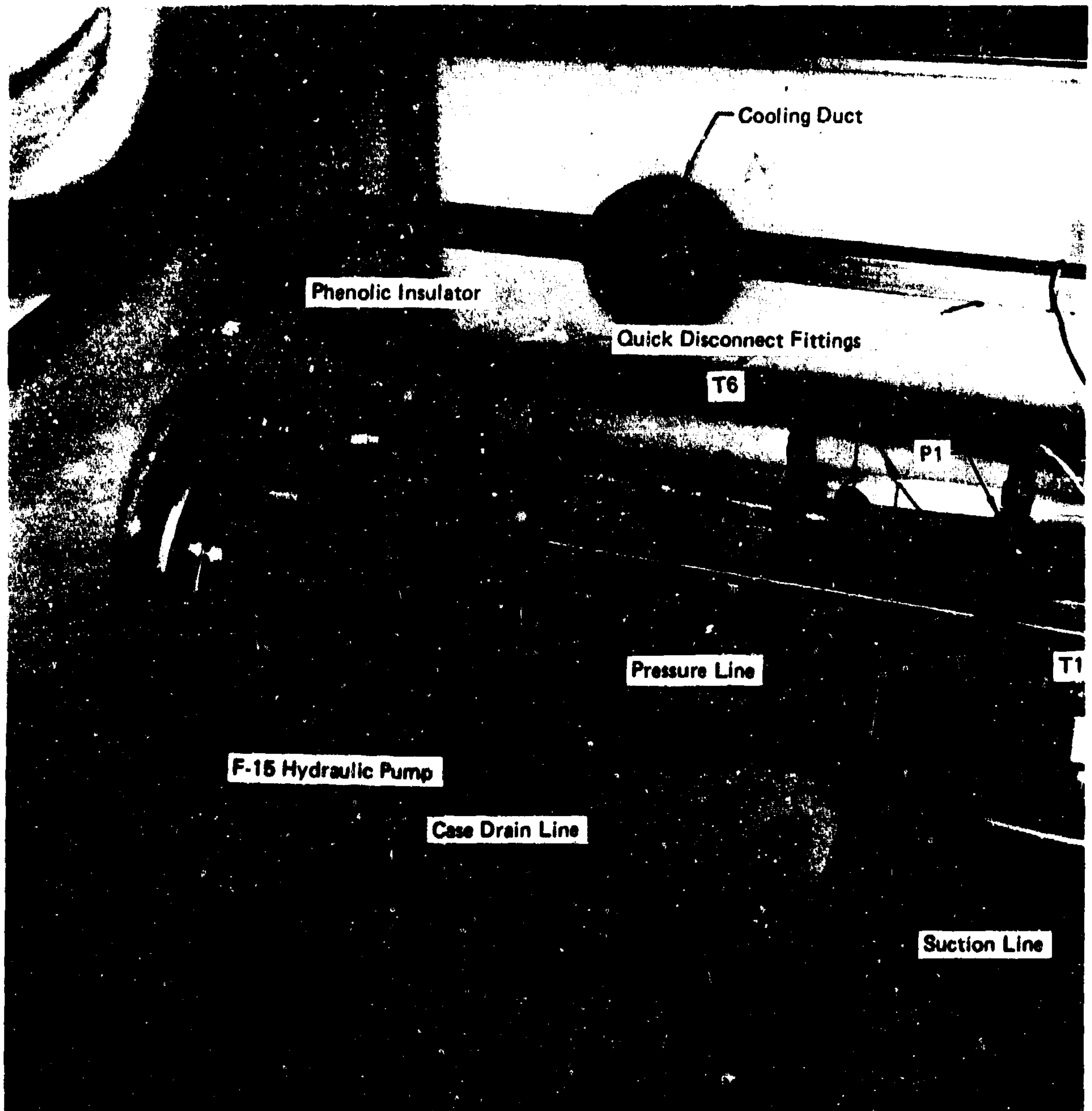

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FIGURE 164 HYTTHA TEST SETUP-PUMP CONFIGURATION

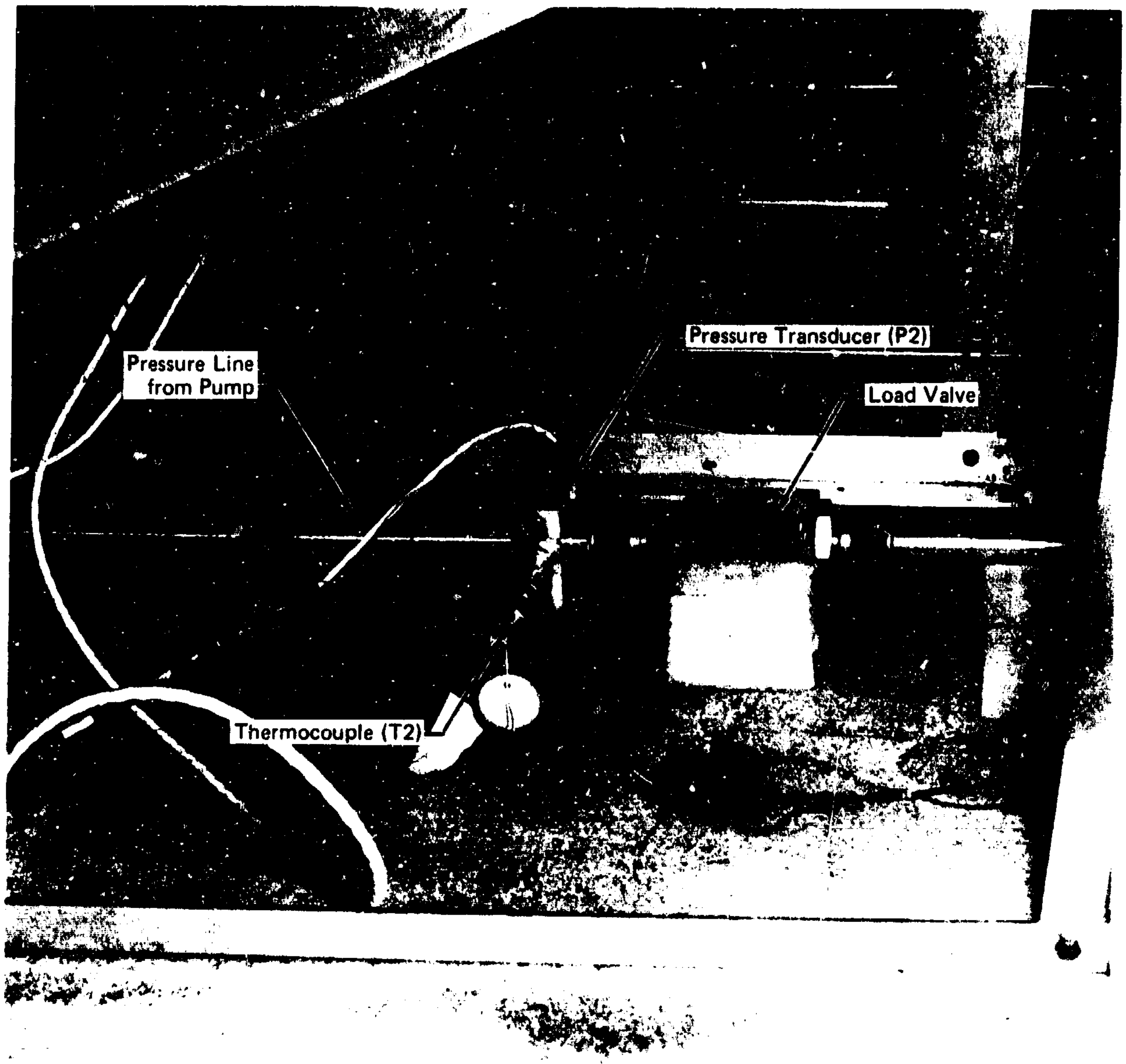


FIGURE 165 HYTTHA TEST SETUP-LOAD CONFIGURATION

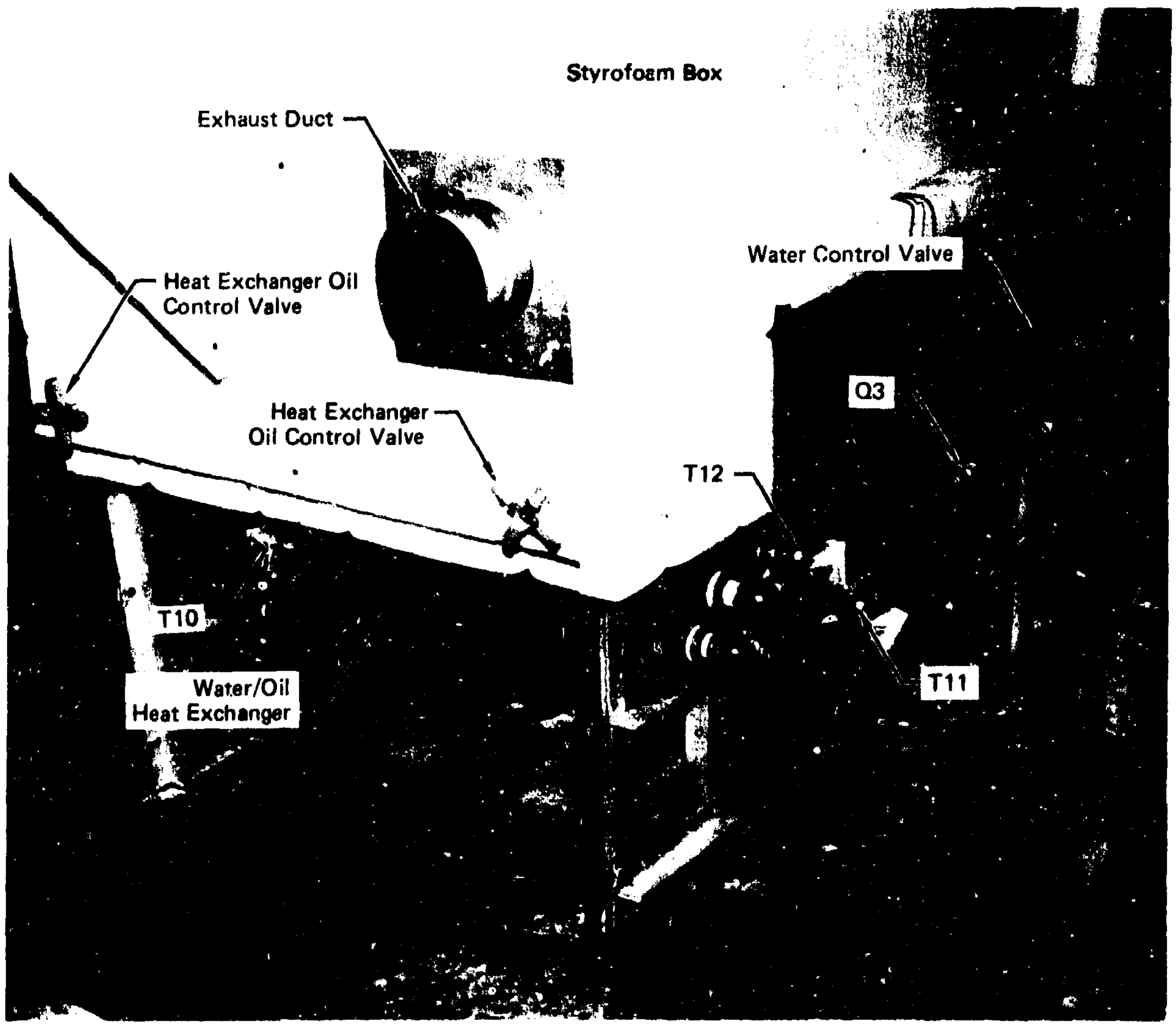


FIGURE 166 HYTTHA TEST SETUP - FLUID COOLING SYSTEM

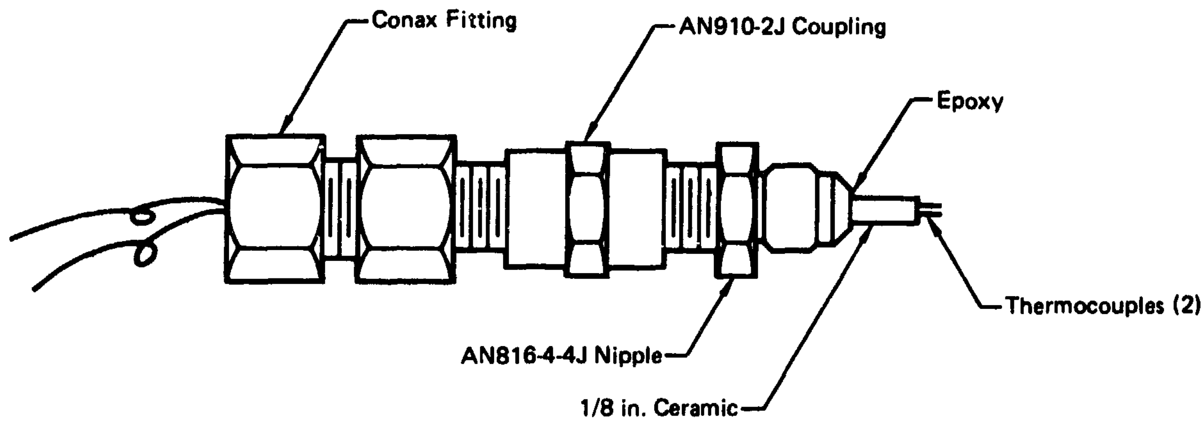


FIGURE 167 THERMOCOUPLE PROBE

Inline thin film strain gage transducers were used to measure pressure. The thermocouples were located next to the pressure transducers and placed 10-15 line diameters downstream from where flows combined in the test system.

Turbine flowmeters measured flows in the case drain, return, and cooling water lines. All data parameters were recorded on analog tape for each run. The data was processed by playback of the analog tape output through waveform recorders and into a calculator. Data was then stored and plotted on digital cassette tapes.

The test setup contained MIL-H-5606C hydraulic fluid and the dissolved air content of the fluid was less than 1% by volume.

b. Test Conditions and Procedures

Testing consisted of steady state and transient thermal response runs. Table 25 contains a listing of the completed tests.

TABLE 25 HYTTHA VERIFICATION TESTS

RUN NUMBER	TEST CONDITIONS	STEADY STATE FLOW (CIS)	SYSTEM ATMOSPHERIC TEMPERATURE (°F)	RESERVOIR PRESSURE (PSIG)	PUMP SPEED (RPM)	TEST RUN TIME
110-1	STEADY STATE TEST	11.55	70.	56	4000	2 MIN 9 SEC
110-2	AFTER TEMPERATURE STABILIZED IN THE SYSTEM, THE HEAT EXCHANGER WAS CLOSED OFF AND THE SYSTEM TEMPERATURE WAS ALLOWED TO INCREASE TO ~200°F	11.55	75.	56-59	4000	13 MIN 17 SEC
110-3	SYSTEM WAS SOAKED AT 70°F THEN PUMP WAS STARTED WITHOUT THE HEAT EXCHANGER IN THE SYSTEM	0-11.55	70.	56-59	4000	10 MIN 32 SEC
110-4	STEADY STATE TEST WITH HEAT EXCHANGER	11.55	0.	54	4000	4 MIN 38 SEC
110-5	SYSTEM WAS COLD SOAKED AT 0°F THEN PUMP WAS STARTED WITHOUT THE HEAT EXCHANGER IN THE SYSTEM	0-11.55	0.	52-54	4000	13 MIN 43 SEC
110-6	STEADY STATE TEST WITHOUT HEAT EXCHANGER SYSTEM WAS ALLOWED TO REACH A STABLE TEMPERATURE BEFORE RECORDING DATA	11.55	0.	56	4000	4 MIN 52 SEC
110-7	SOAKED SYSTEM AT 70°F, THEN STARTED THE PUMP WITHOUT THE HEAT EXCHANGER IN THE SYSTEM	115.5	70.	62	4000	4 MIN 16 SEC

For the steady state tests the system ran at a stabilized temperature and flow,and the fluid and component temperatures were recorded. The heat exchanger cooling flow rate and temperature were also recorded.

Transient testing investigated start-ups and load heat generation. The test stand was started from cold soaked and room temperature conditions. The subsequent transient thermal response was recorded. In addition the system was stabilized at a constant operating temperature and the heat exchanger was closed to allow the load to warm the system.

The F-15 pump locked after completing run 110-7 and the pumps spline drive shaft sheared. The following temperatures were recorded prior to the pump failure.

	Temperature (°F)
Pump Case Drain (T5)	213.
Pump Inlet (T6)	277.
Pump Case (T8)	217.
Upstream Of Load Valve (T2)	267.
Downstream of Load Valve (T3)	284.

When the pump was disassembled, several large pieces of steel believed to be from a piston shoe hold down ring were found in the pump housing. Fine bronze particles were also found. Testing was terminated due to limited time and budget available to clean out the system and complete the test series.

The F-15 hydraulic pump (ABEX P/N 69796-12, S/N 105, USAF ID 263053) was a qualification unit used for vibration and shock testing on the F-15 program. In addition the unit was subjected to proof pressure and overspeed tests.

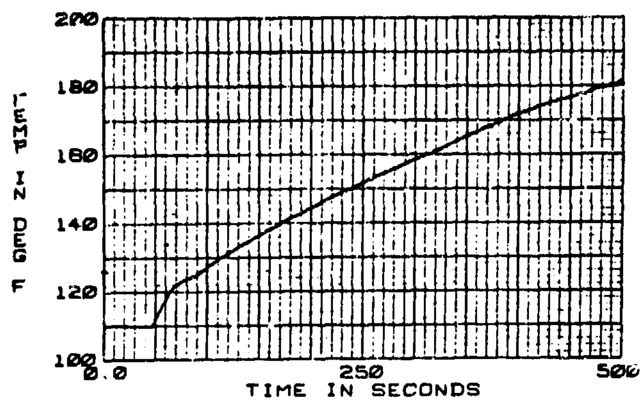
c. Test Results and Program Verification

Test data for the heat exchanger off transient (Run No. 110-2) is shown in Figure 168. The ambient temperature was 75°F and the heat exchanger was used to stabilize the system temperatures. To generate the thermal transient the bypass valve shown in Figure 163 was opened and the heat exchanger valves were closed.

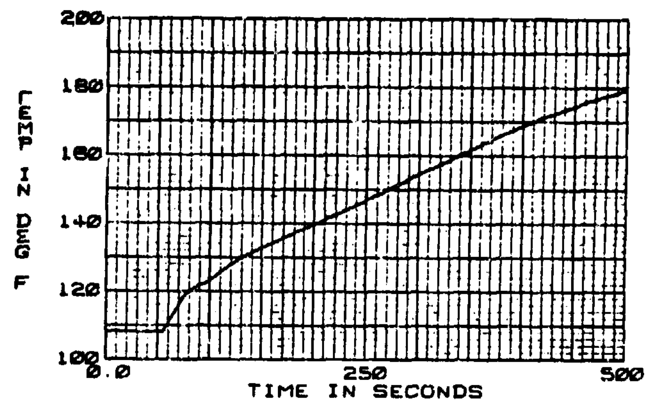
A computer simulation was made of the test run. The equivalent HYTTHA computer schematic of the test system is shown in Figure 169. The steady state leg and node diagram is presented in Figure 170. The HYTTHA input data is in Table 26. The simulated pump outlet fluid temperature is shown in Figure 171. The computed fluid temperature rises too quickly. The computed value at 500 seconds is 30°F above the actual data.

To determine if the pump model was causing the rapid fluid temperature rise, the pump was replaced by a constant pressure reservoir with the pump outlet fluid temperature profile. The results of the simulation in Figures 172 and 173 show better correlation with the test data.

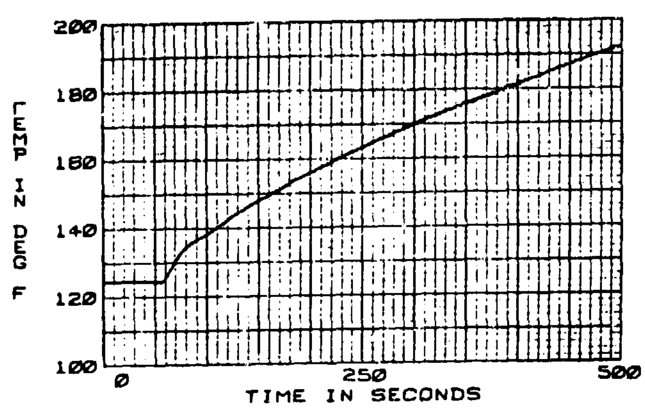
Test data for the start-up thermal transient data run 110-5 is shown in Figure 174. A computer simulation was made of the start-up transient run. The HYTTHA input data file is presented in Table 27. The computer output runs are shown in Figures 175 and 176. The HYTTHA program predicts a 14°F higher temperature for the start-up pump outlet fluid temperature peak at 10 seconds into the simulation. The computed fluid temperature at 200 seconds is 20°F higher than the measured value. The computed and actual fluid temperatures downstream of the pump outlet are shown in Figure 176.



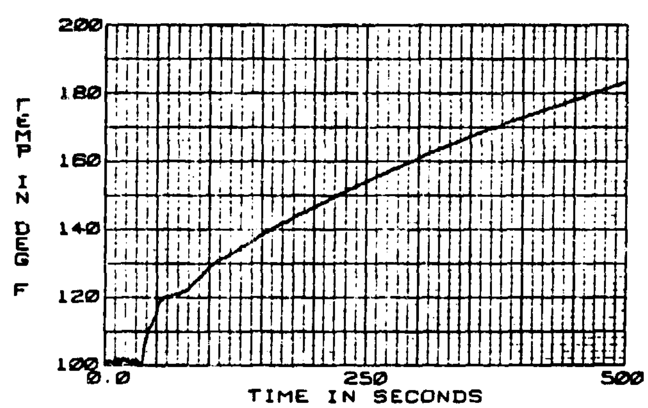
THERMAL TRANSIENT TEST
110-2-T1 HEAT EXCHANGER OFF TRANSIENT
11.55 CIS



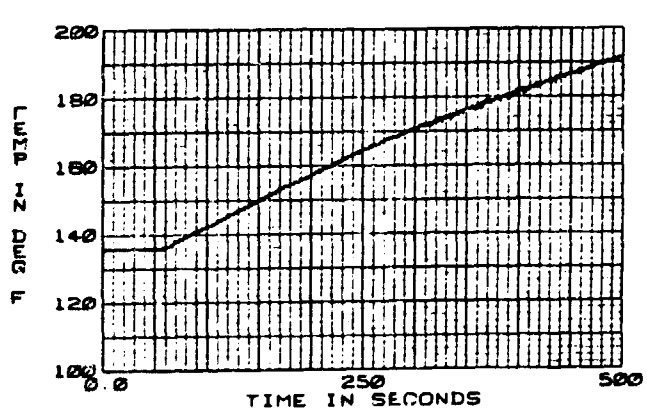
THERMAL TRANSIENT TEST
110-2-T2 HEAT EXCHANGER OFF TRANSIENT
11.55 CIS



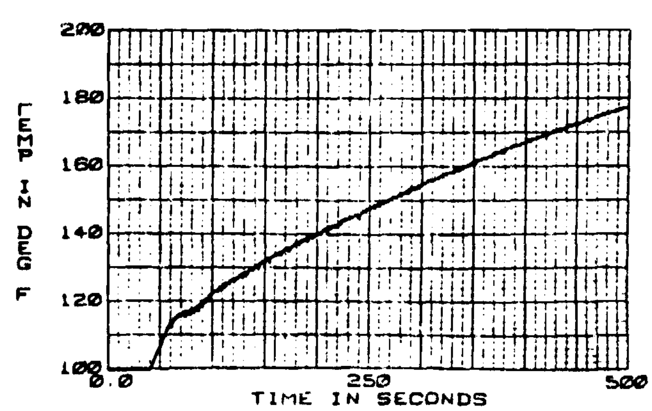
THERMAL TRANSIENT TEST
110-2-T3 HEAT EXCHANGER OFF TRANSIENT
11.55 CIS



THERMAL TRANSIENT TEST
110-2-T4 HEAT EXCHANGER OFF TRANSIENT
11.55 CIS

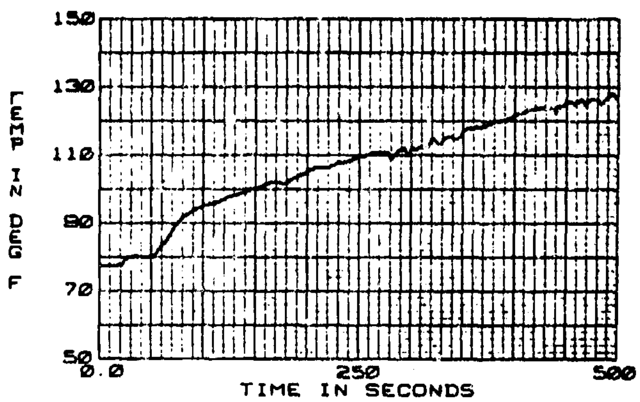


THERMAL TRANSIENT TEST
110-2-T5 HEAT EXCHANGER OFF TRANSIENT
11.55 CIS

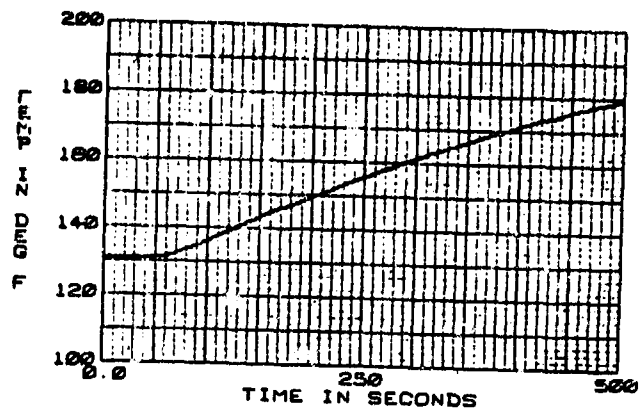


THERMAL TRANSIENT TEST
110-2-T6 HEAT EXCHANGER OFF TRANSIENT
11.55 CIS

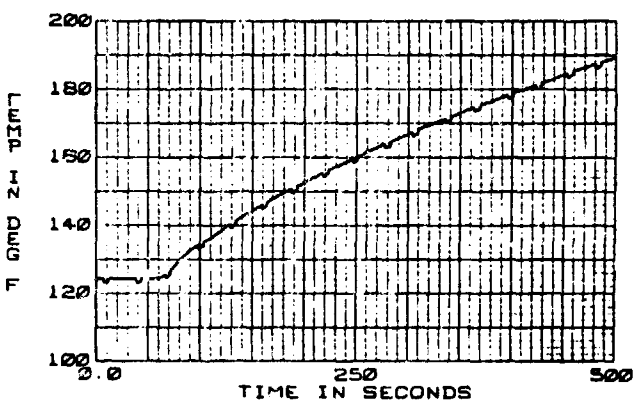
FIGURE 168 DATA RUN NUMBER 110-2



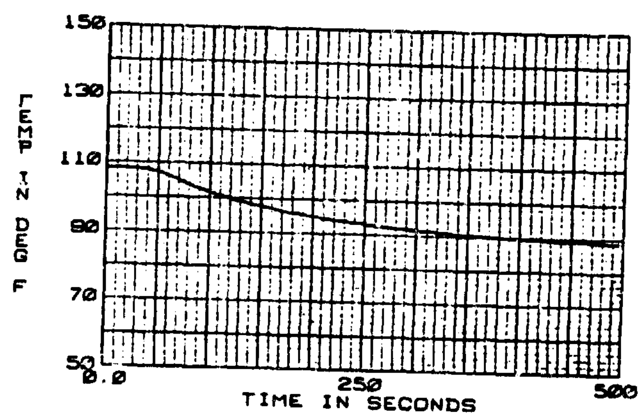
THERMAL TRANSIENT TEST
110-2-T7 HEAT EXCHANGER OFF TRANSIENT
11.55 CIS



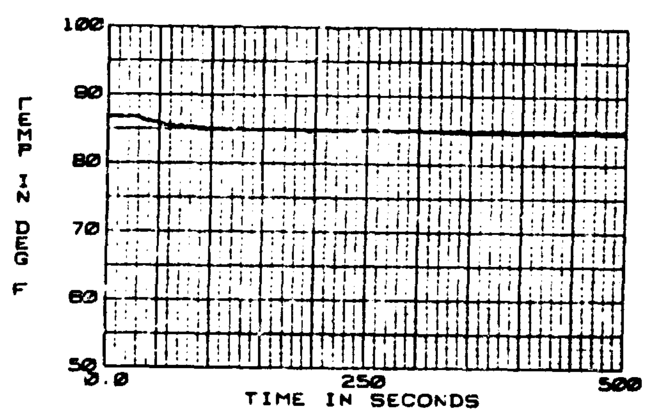
THERMAL TRANSIENT TEST
110-2-T8 HEAT EXCHANGER OFF TRANSIENT
11.55 CIS



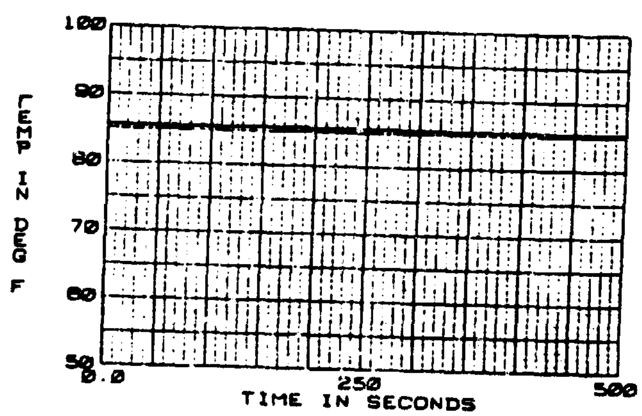
THERMAL TRANSIENT TEST
110-2-T9 HEAT EXCHANGER OFF TRANSIENT
11.55 CIS



THERMAL TRANSIENT TEST
110-2-T10 HEAT EXCHANGER OFF TRANSIENT
11.55 CIS

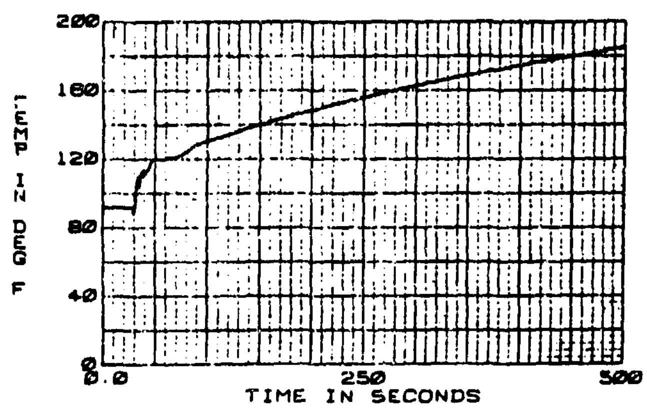


THERMAL TRANSIENT TEST
110-2-T11 HEAT EXCHANGER OFF TRANSIENT
11.55 CIS

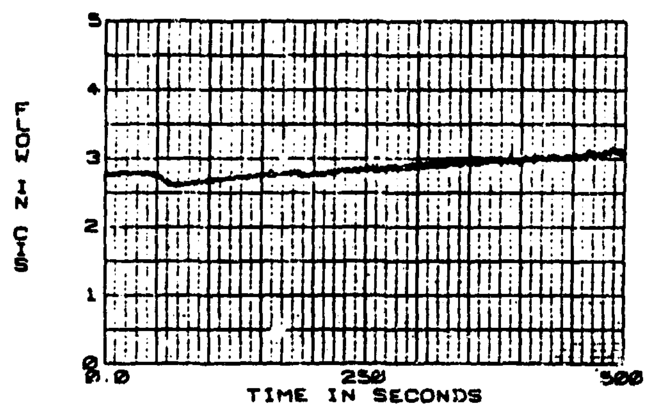


THERMAL TRANSIENT TEST
110-2-T12 HEAT EXCHANGER OFF TRANSIENT
11.55 CIS

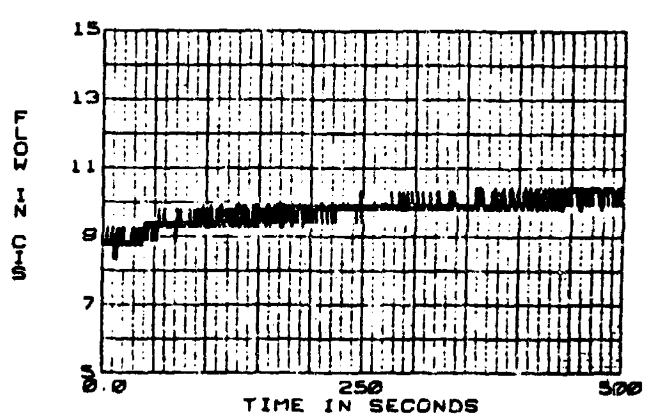
FIGURE 168 DATA RUN NUMBER 110-2 (CONTINUED)



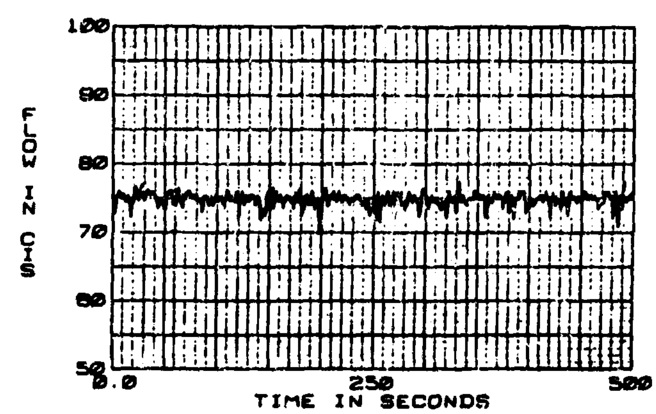
THERMAL TRANSIENT TEST
110-2-T13 HEAT EXCHANGER OFF TRANSIENT
11.55 CIS



THERMAL TRANSIENT TEST
110-2-Q1 HEAT EXCHANGER OFF TRANSIENT
11.55 CIS



THERMAL TRANSIENT TEST
110-2-Q2 HEAT EXCHANGER OFF TRANSIENT
11.55 CIS



THERMAL TRANSIENT TEST
110-2-Q3 HEAT EXCHANGER OFF TRANSIENT
11.55 CIS

FIGURE 168 DATA RUN NUMBER 110-2 (CONTINUED)

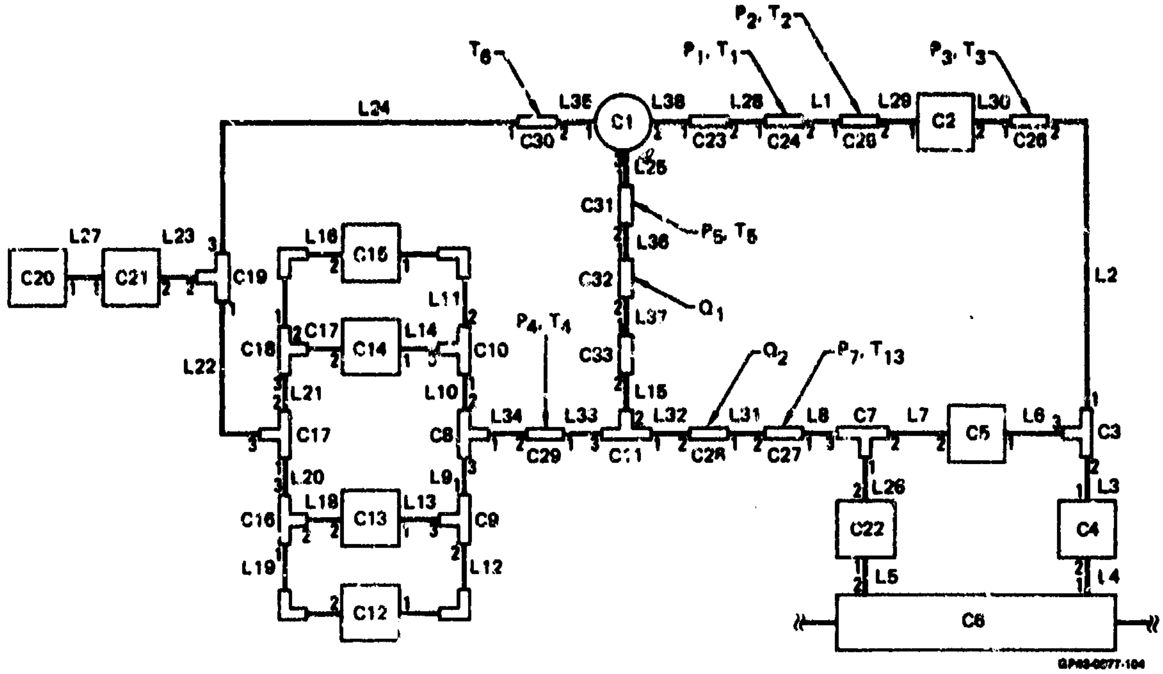


FIGURE 169 HYTTHA TEST VERIFICATION SYSTEM COMPUTER SCHEMATIC

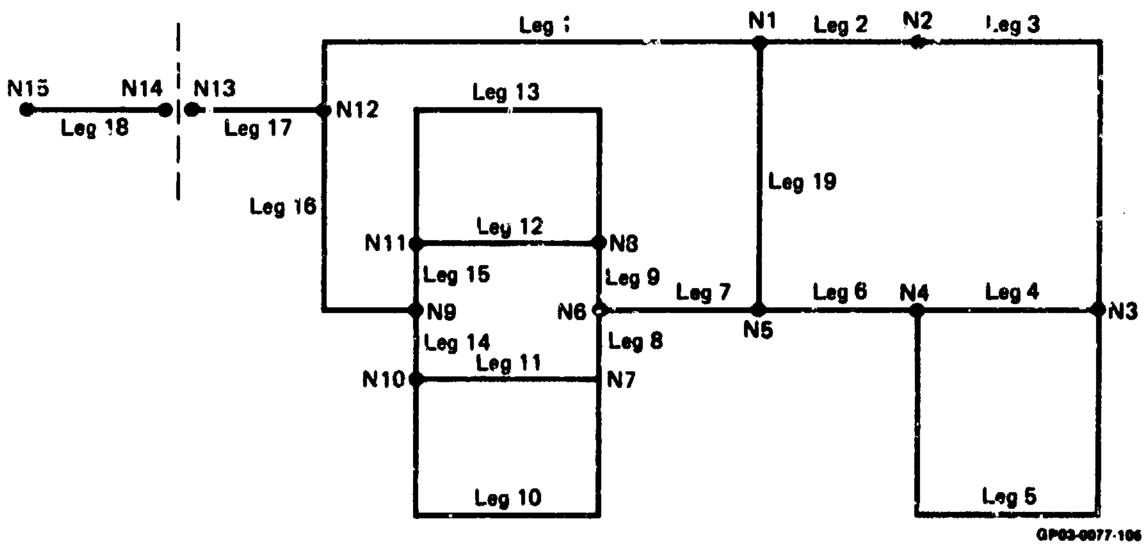


FIGURE 170 HYTTHA TEST VERIFICATION SYSTEM STEADY STATE LEG AND NODE DIAGRAM

TABLE 26 TEST RUN NO. 110-2 EXCHANGER OFF TRANSIENT HYTTHA INPUT DATA

```
** HYTTHA TEST RUN NO. "110-2"  -- HEAT EXCHANGER OFF TRANSIENT -- (DTHERM) **
 38   33          .5       500.        5.
  1        80.          2.       14.7
  1    9    0    0    2   90   90
   201.5         1.      .0580           4.      .0069       80.       70.      111.
  2    9    0    0    3   90   90   90
 180.000         1.      .0580           4.      .0069       70.       70.      126.
  3    9    0    0    0    0    0    0
    2.75         1.      .0580           4.      .0069       70.       70.      126.
  4    9    0    0    0    0    0    0
      2.         1.      .0580           4.      .0069       80.       70.      126.
  5    9    0    0    0    0    0    0
      2.         1.      .0580           4.      .0069       80.       70.       92.
  6    9    0    0    0    0    0    0
   11.25         1.      .0580           4.      .0069       70.       70.      115.
  7    9    0    0    0    0    0    0
     11.         1.      .0580           4.      .0069       70.       70.      105.
  8    9    0    0    0    0    0    0
    25.5         1.      .0560           4.      .0069       70.       70.       92.
  9    9    0    0    0    0    0    0
     7.0        .75      .0420           4.      .0069       70.       70.      101.
 10    9    0    0    0    0    0    0
     7.0        .75      .0420           4.      .0069       70.       70.      101.
 11    9    0    0    1   90
   7.375        .75      .0420           4.      .0069       70.       70.      101.
 12    9    0    0    1   90
   7.375        .75      .0420           4.      .0069       70.       70.      101.
 13    9    0    0    0    0    0    0
      2.        .75      .0420           4.      .0069       70.       70.      101.
 14    9    0    0    0    0    0    0
      2.        .75      .0420           4.      .0069       70.       70.      101.
 15    9    0    0    0    0   90    0
  78.375        375      .0220           4.      .0069       70.       70.      136.
 16    9    0    0    1   90
   7.375        .75      .0420           4.      .0069       70.       70.      101.
 17    9    0    0    0    0    0    0
      2.        .75      .0420           4.      .0069       70.       70.      101.
 18    9
      2.        .75      .0420           4.      .0069       70.       70.      101.
 19    9    0    0    1   90
   7.375        .75      .0420           4.      .0069       70.       70.      101.
 20    9
      7.        .75      .0420           4.      .0069       70.       70.      101.
 21    9    0    0    0    0
      7.        .75      .0420           4.      .0069       70.       70.      101.
 22    9    0    0    2   90   90
  40.875         1.      .0580           4.      .0069       70.       70.      101.
 23    9
   3.125         1.      .0580           4.      .0069       70.       70.       85.
 24    9    0    0    1   90
  28.625         1.      .0580           4.      .0069       70.       70.      100.
 25    9    0    0    2   90   90
  29.250       .375      .0220           4.      .0069      100.       70.      136.
 26    9
    2.75         1.      .0580           4.      .0069       70.       70.       92
```

TABLE 26 TEST RUN NO. 110-2 EXCHANGER OFF TRANSIENT HYTTHA INPUT DATA (CONTINUED)

27 9							
10.375	1.	.0580	4.	.0069	70.	70.	70.
28 9							
15.625	1.	.0580	4.	.0069	90.	70.	111.
29 9							
1.625	1.	.0580	4.	.0069	75.	70.	108.
30 9							
15.125	1.	.0580	4.	.0069	75.	70.	126.
31 9	0 0	1 135					
37.	1.	.0580	4.	.0069	70.	70.	94.
32 9	0 0	1 135					
6.625	1.	.0580	4.	.0069	70.	70.	96.
33 9							
15.25	1.	.0580	4.	.0069	70.	70.	98.
34 9							
15.125	1.	.0580	4.	.0069	70.	70.	101.
35 9							
10.	1	.0580	4.	.0069	100.	70.	100.
36 9							
15.125	.375	.0220	4.	.0069	70.	70.	136.
37 9							
1.625	.375	.0220	4.	.0069	70.	70.	136.
38 9							
4.875	1.2	0.1	4.	.0069	100.	70.	111.
1 51	4 35	-38 -25	1	2 19			
9.	9.	5.716	8.155	8.529	15.	10.	175.
8.	3.	.63	5675.	6.	27.	69.	.0069
.69	.189	100.	70.	100.	131.	220.	4600.
4015.	3050.	2850.	25.	35.	8.2	50.	8.
2 41	2 29	-30					
9.	1.661	1.	5.25	15.	1.0E-10	1.	75.
70.	108.	110.	.65	.0513			
3 11	2 2	-3 -6					
9.	.2	.69	2.3	5.9	5.1	.0069	70.
70.	115.	110.					
4 21	5 3	-4 4					
9.	9.	2.	.1	3.	.3	2.5	12.
6.6	.033	.0069	1.	85.	85.	126.	120.
.022	.65						
0.0	25.0	27.0	500.				
5.0	5.0	0.0	0 0				
5 21	5 6	-7 4					
9.	9.	2.145	.1	3.	.3	2.5	12.
6.6	.033	.0069	1.	70.	70.	70.	70.
.022	.65						
0.0	24.0	24.5	500.				
0.0	0.0	5.0	5.0				
6 69	3 4	-5					
9.		95.	45.	510.	1.	41.	400.
4000.	900.	4896.	.0069	.069	.075	2.91	85.
85.	85.	126.	92.	87.	.4		5.
7 11	2 26	7 -8					
9.	.2	.69	.9592	5.9	5.1	.0069	70.
70.	92.	92.					
8 11	2 34	-10 -9					

TABLE 26 TEST RUN NO. 110-2 EXCHANGER OFF TRANSIENT HYTTHA INPUT DATA (CONTINUED)

	9.		.2		.69	1.578	5.9	5.1	0069	70.
	70.		101.		101.					
9	11	2	9	-12	-13					
	9.		.2		.69	2.3	5.9	5.1	.0069	70.
	70.		101.		101.					
10	11	2	10	-11	-14					
	9.		.2		.69	2.3	5.9	5.1	.0069	70.
	70.		101.		101.					
11	11	2	32	15	-33					
	9.		.2		.69	2.3	5.9	5.1	.0069	70.
	70.		101.		101.					
12	81	2	12	-19						
	9.		4.627		10.	.083	9.63	99.	69.	.0069
	1.		70.		70.	101.	90.	.01	.0002	
13	81	2	13	-18						
	9.		4.979		10.	.083	9.63	99.	69.	.0069
	1.		70.		70.	101.	90.	.01	.0002	
14	81	2	14	-17						
	9.		4.979		10.	.083	9.63	99.	69.	.0069
	1.		70.		70.	101.	90.	.01	.0002	
15	81	2	11	-16						
	9.		4.627		10.	.083	9.63	99.	69.	.0069
	1.		70.		70.	101.	90.	.01	.0002	
16	11	2	19	18	-20					
	9.		.2		.69	2.3	5.9	5.1	.0069	70.
	70.		101.		101.					
17	11	2	20	21	-22					
	9.		.2		.69	1.578	5.9	5.1	.0069	70.
	70.		101		101.					
18	11	2	16	17	-21					
	9.		.2		.69	2.3	5.9	5.1	.0069	70.
	70.		101.		101.					
19	11	2	22	23	-24					
	9.		.2		.69	1.052	5.9	5.1	.0069	70.
	70.		101.		101.					
20	61	1	-27							
	3600.		70.		70.					
21	62	3	27	-23						
	9.		9.		17.36	93.410	1.558	4.67	5.467	1.75
	10.992		.11		665.	155.	.0069	.06	3.5	10.
	1.		70.		70.	77.	75.			
22	21	5	5	-26	4					
	9.		9.		2.	.1	3.	.3	2.5	12.
	6.6		.033		.0069	1.	85.	85.	92.	92.
	.022		.65							
	0.0		25.0		27.0	500.				
	5.0		5.0		0.0	0.0				
23	11	2	38	-28						
	9.		0.001		0.001	0.001	0.001	0.001	.0069	120.
	70.		120.		120.					
24	11	2	28	-1						
	4.		0.32146		0.61375	.8646	14.429	2.7772	.0069	70.
	70.		111.		110.					
25	11	2	1	-29						
	4.		0.32146		0.61375	1.	14.429	2.7772	.0069	90.

TABLE 26 TEST RUN NO. 110-2 HEAT EXCHANGER OFF TRANSIENT HYTTHA INPUT DATA (CONTINUED)

```
      70.     108.     108.
 26   11    2   30   -2
       4.  0.32146  0.61375      .5    14.429    2.7772     .0069     75.
      70.     126.     126.
 27   11    2    8  -31
       4.    .5214  0.61375      1.    14.429    2.7772     .0069     70.
      70.      92.      92.
 28   11    2   31  -32
       4.    1.217  0.61375      1.    14.429    2.7772     .0069     70.
      70.      92.      92.
 29   11    2   33  -34
       4.  0.32146  0.61375      1.    14.429    2.7772     .0069     70.
      70.     101.     101.
 30   11    2   24  -35
       4.  0.32146  0.61375      1.    14.429    2.7772     .0069     70.
      70.     100.     100.
 31   11    2   25  -36
       4.  0.32146  0.61375   .5742    14.429    2.7772     .0069    100.
      70.     136.     136.
 32   11    2   36  -37
       4.  0.32146  0.61375   .3080    14.429    2.7772     .0069     70.
      70.     136.     136.
 33   11    2   37  -15
       9.    .0726      .69      1.       5 9       5.1     .0069     70.
      70.     136.     136.
 15   19   01
  1   12    1    5         16.
 19    3    0   24   30     1    0   35    1    1
  2    1    2    8         12.
  1    2    0   38   23     1    0   28   24    1    0    1   25    1    0   29
  3    2    3    5         12.
  2    1    0   30   26     1    0    2    3    1
  4    3    4    5          0.
  3    3    0    6    5     1    0    7    7    2
  5    3    4    9         12.
  3    2    0    3    4     1    0    4    6    1    0    5   22    1    0   26
  7    1
  6    4    5    7         12.
  7    3    0    8   27     1    0   31   28    1    0   32   11    1
  7    5    6    5         16.
 11    3    0   33   29     1    0   34    8    1
  8    6    7    3          8.
  8    3    0    9    9     1
  9    6    8    3          8.
  8    2    0   10   10     1
 10    7   10    5          4.
  9    2    0   12   12     1    0   19   16    1
 11    7   10    5          4.
  9    3    0   13   13     1    0   18   16    2
 12    8   11    5          4.
 10    3    0   14   14     1    0   17   18    2
 13    8   11    5          4.
 10    2    0   11   15     1    0   16   18    1
 14   10    9    3          8.
 16    3    0   20   17     1
 15   11    9    3          8.
 18    3    0   21   17     2
 16    9   12    3         16.
 17    3    0   22   19     1
 17   13   12    3         .01
 21    2    0   23   19     2
 18   15   14    3         .01
 20    1    0   27   21     1
 19    1    5    9          4.
  1    3    0   25   31     1    0   36   32    1    0   37   33    1    0   15
 11    2
  7   18    0   00  000
  1    4    1    4   -1    -4
  2    2    1    4
 23    2   -2   -4
 31    3    1    4   -2
 33    2   -1   -4
 35    1    4
 36    3    1    4   -2
  1    1    1    2    1     3    1    4    1    5    1    6    1    7    1    8
  1    9    1   10    1    11    1   12    1   13    1   14    1   15    1   16
  1   17    1   18
```

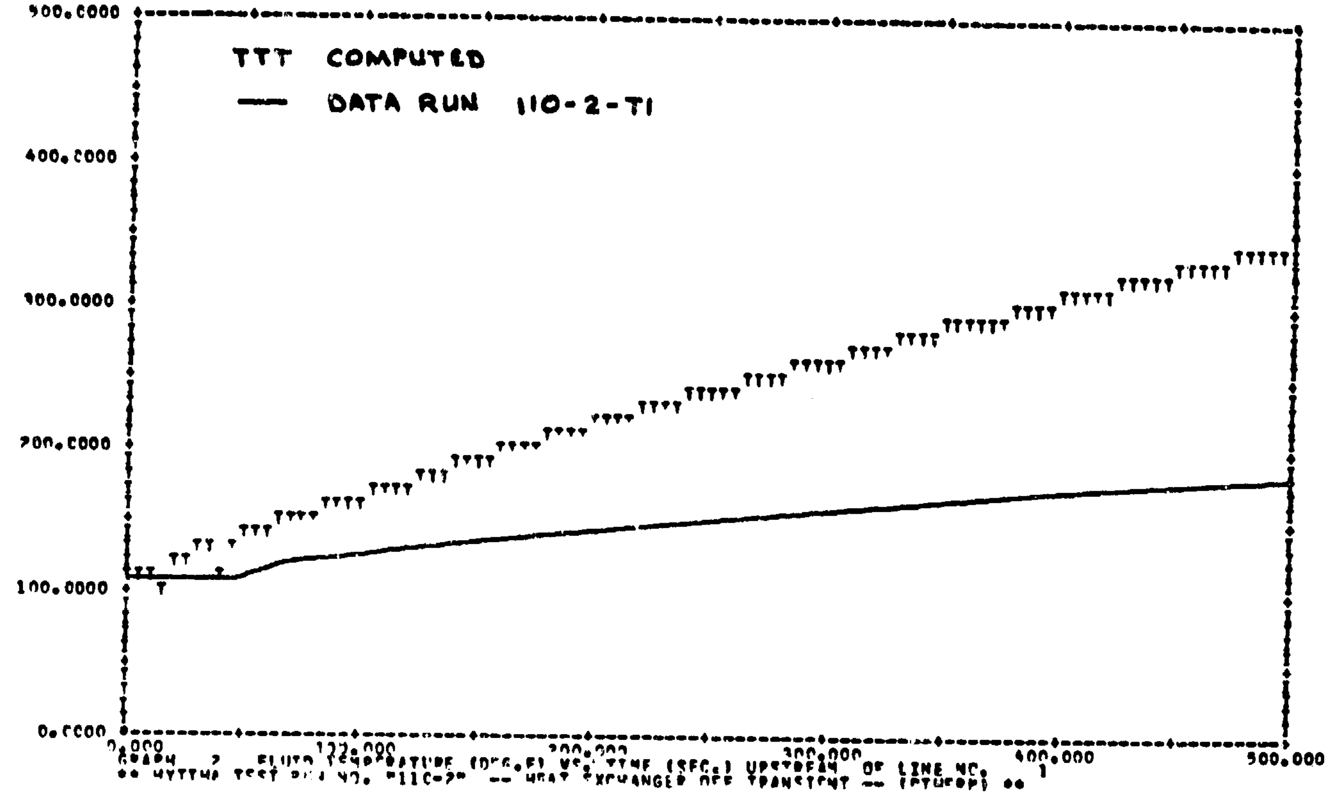


FIGURE 171 HYTTHA SIMULATION DATA RUN 110-2-T1

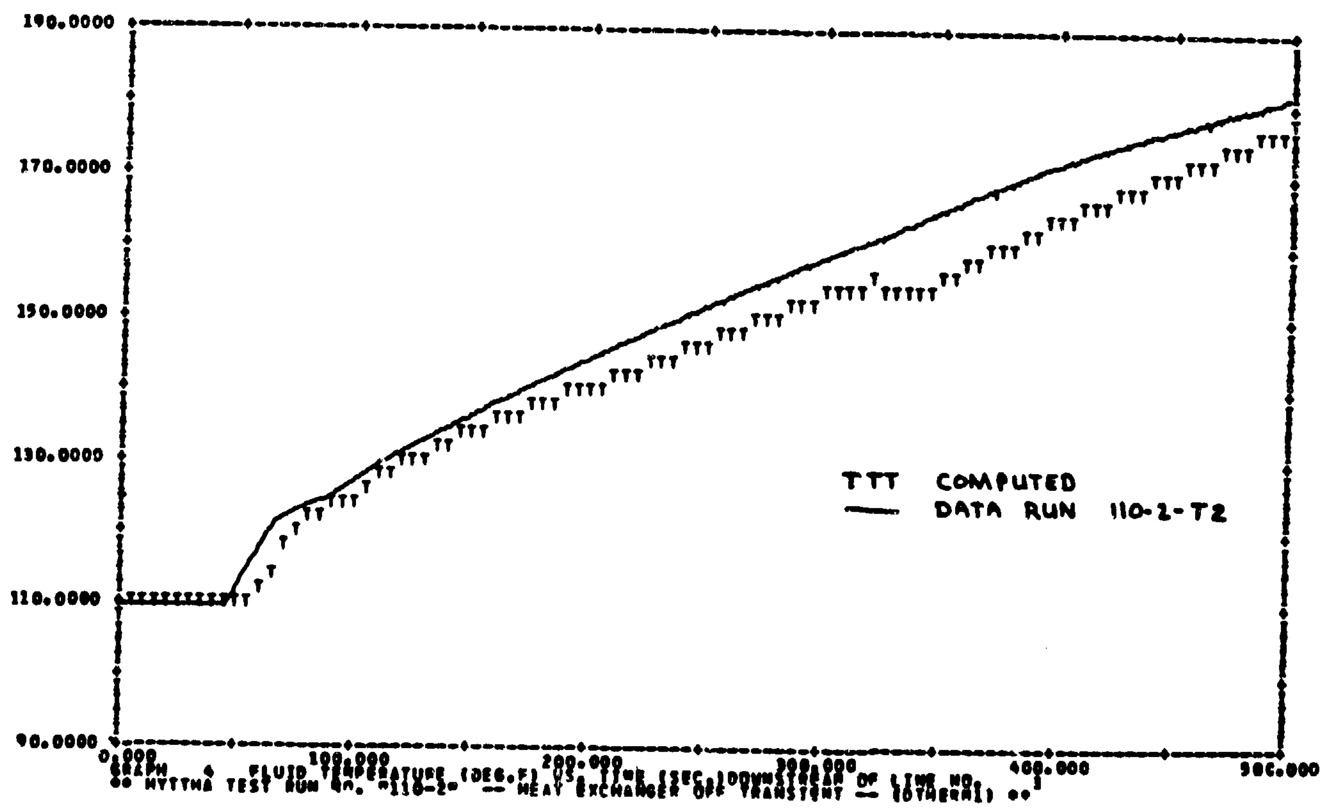


FIGURE 172 HYTTHA SIMULATION DATA RUN 110-2-T2

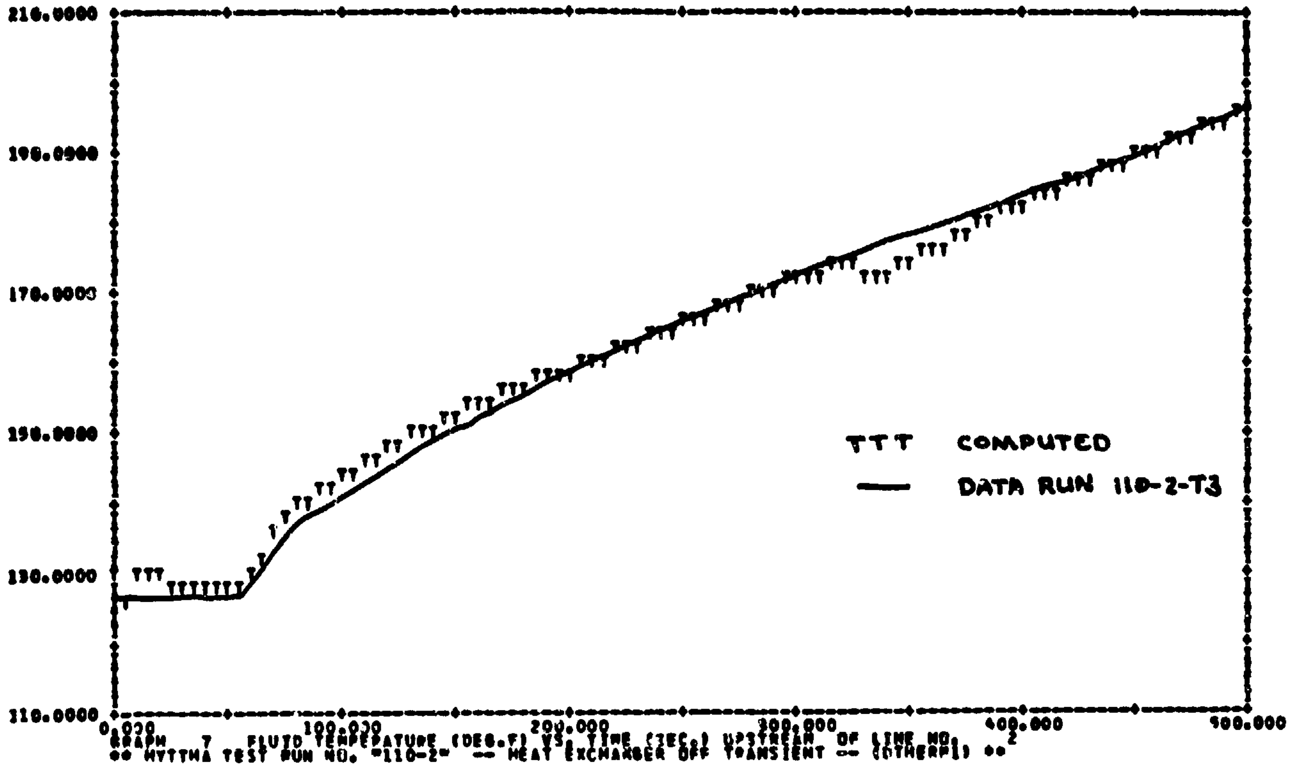


FIGURE 173 HYTTHA SIMULATION DATA RUN 110-2-T3

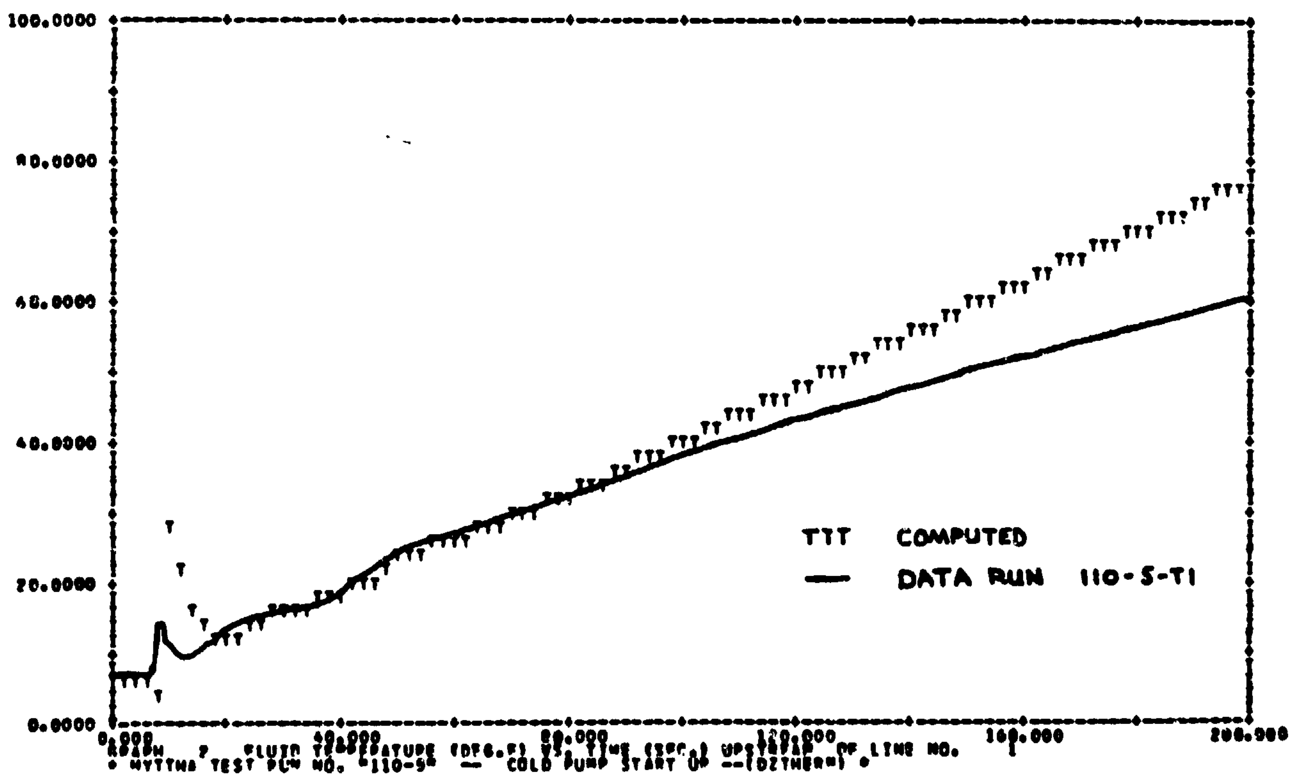


FIGURE 175 HYTTHA SIMULATION DATA RUN 11C-5-T1

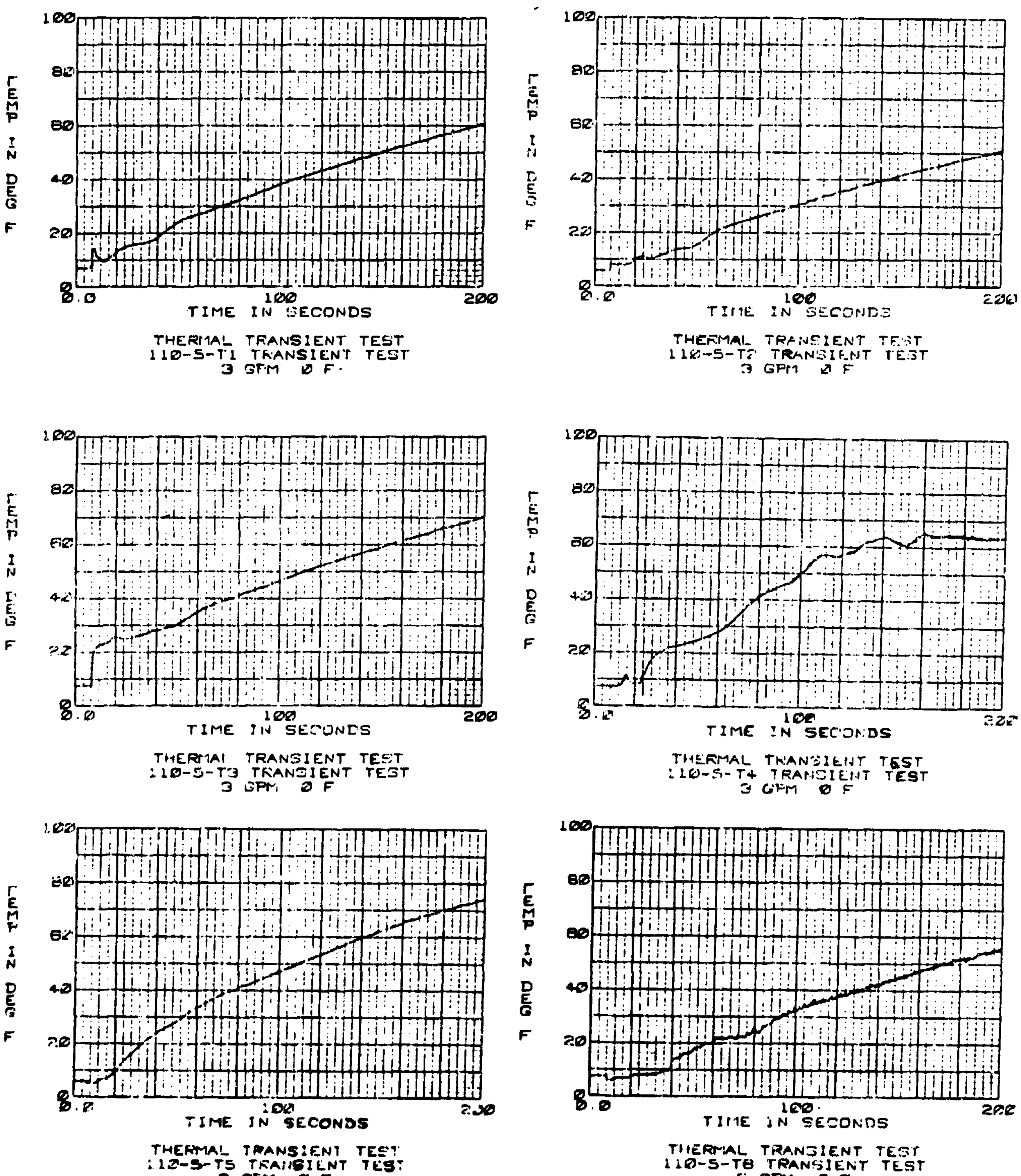
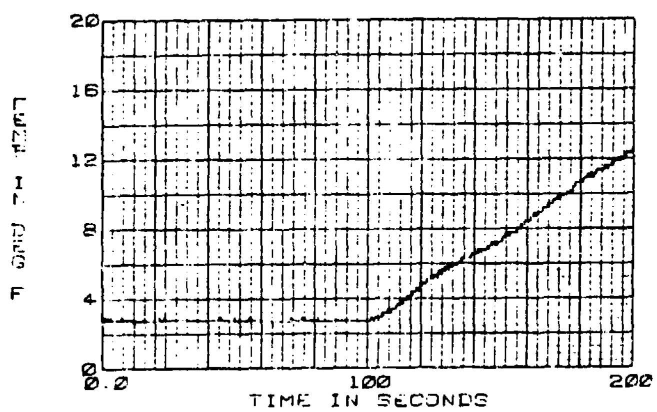
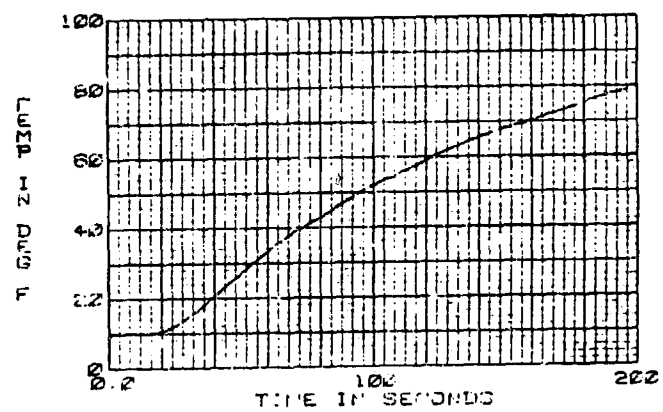


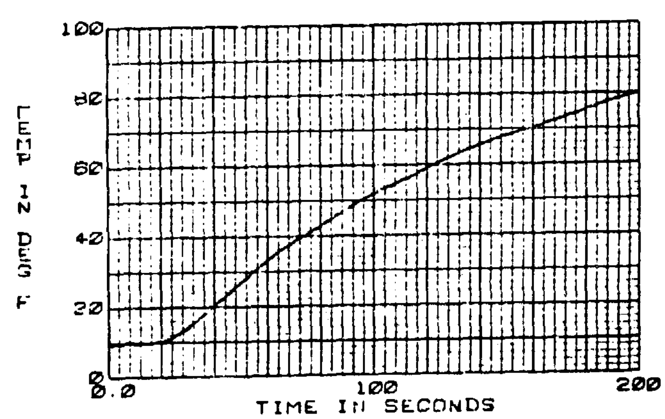
FIGURE 174 DATA RUN NUMBER 110-5



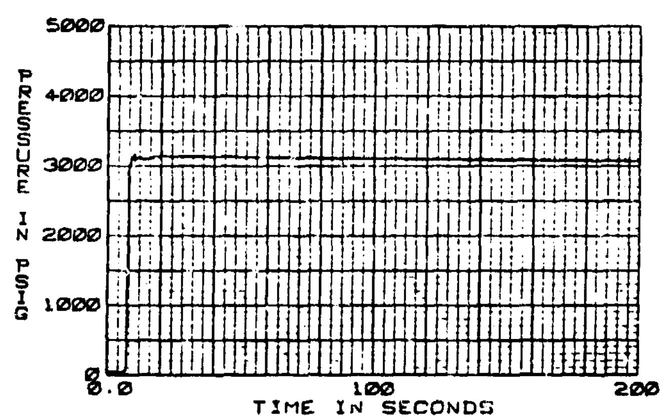
THERMAL TRANSIENT TEST
110-5-T7 TRANSIENT TEST
3 GPM 0 F



THERMAL TRANSIENT TEST
110-5-T8 TRANSIENT TEST
3 GPM 0 F

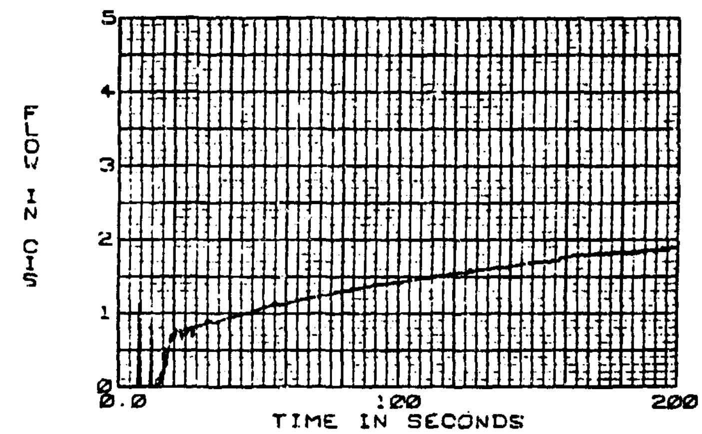


THERMAL TRANSIENT TEST
110-5-T9 TRANSIENT TEST
3 GPM 0 F

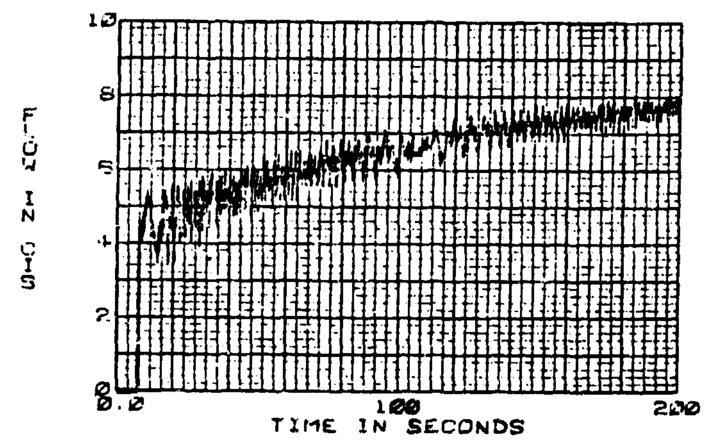


THERMAL TRANSIENT TEST
110-5-P1 PUMP START UP TRANSIENT
3 GPM 0 F

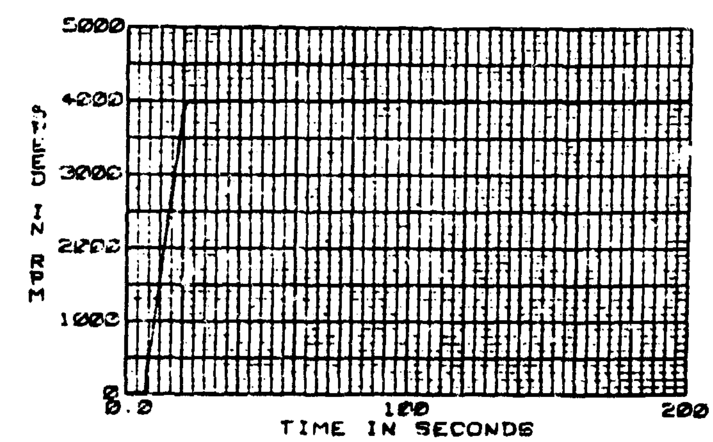
FIGURE 174 DATA RUN NUMBER 110-5 (CONTINUED)



THERMAL TRANSIENT TEST
110-5-Q1 TRANSIENT TEST
3 GPM 0 F



THERMAL TRANSIENT TEST
110-5-Q2 TRANSIENT TEST
3 GPM 0 F



THERMAL TRANSIENT TEST
110-5-DS TRANSIENT TEST
3 GPM 0 F

FIGURE 174 DATA RUN NUMBER 110-5 (CONTINUED)

TABLE 27 TEST RUN NO. 110-5 COLD PUMP START-UP
HYTTHA INPUT DATA

```
* HYTTHA TEST RUN NO. "110-5"  --  COLD PUMP START UP --(DZTHERM) *
 38   33        .5      200.           2.
  1        6.        2.      14.7
  1    9   .0    0    2   90    90
   201.5         1.     .0580          4.     .0069      0.       6.       6.
  2    9    0    0    3   90    90    90
   180.0         1.     .0580          4.     .0069      0.       6.       6.
  3    9    0    0    0    0     0     0
    2.75         1.     .0580          4.     .0069      0.       6.       6.
  4    9    0    0    0    0     0     0
      2.         1.     .0580          4.     .0069      0.       6.       6.
  5    9    0    0    0    0     0     0
      2.         1.     .0580          4.     .0069      0.       6.       6.
  6    9    0    0    0    0     0     0
   11.25         1.     .0580          4.     .0069      0.       6.       6.
  7    9    0    0    0    0     0     0
     11.         1.     .0580          4.     .0069      0.       6.       6.
  8    9    0    0    0    0     0     0
    25.5         1.     .0580          4.     .0069      0.       6.       6.
  9    9    0    0    0    0     0     0
     7.0        .75     .0420          4.     .0069      0.       6.       6.
 10    9    0    0    0    0     0     0
     7.0        .75     .0420          4.     .0069      0.       6.       6.
 11    9    0    0    1   90
   7.375        .75     .0420          4.     .0069      0.       6.       6.
 12    9    0    0    1   90
   7.375        .75     .0420          4.     .0069      0.       6.       6.
 13    9    0    0    0    0     0     0
      2.        .75     .0420          4.     .0069      0.       6.       6.
 14    9    0    0    0    0     0     0
      2.        .75     .0420          4.     .0069      0.       6.       6.
 15    9    0    0    1   90
  78.375       .375     .0220          4.     .0069      0.       6.       6.
 16    9    0    0    1   90
   7.375        .75     .0420          4.     .0069      0.       6.       6.
 17    9    0    0    0    0     0     0
      2.        .75     .0420          4.     .0069      0.       6.       6.
 18    9
      2.        .75     .0420          4.     .0069      0.       6.       6.
 19    9    0    0    1   90
   7.375        .75     .0420          4.     .0069      0.       6.       6.
 20    9
      7.        .75     .0420          4.     .0069      0.       6.       6.
 21    9    0    0    0    0
      7.        .75     .0420          4.     .0069      0.       6.       6.
 22    9    0    0    2   90    90
  40.875         1.     .0580          4.     .0069      0.       6.       6.
 23    9
  3.1250         1.     .0580          4.     .0069      0.       6.       6.
 24    9    0    0    1   90
  28.625         1.     .0580          4.     .0069      0.       6.       6.
 25    9    0    0    2   90    90
   29.25       .375     .0220          4.     .0069      0.       6.       6.
 26    9
    2.75         1.     .0580          4.     .0069      0.       6.       6.
```

TABLE 27 TEST RUN NO. 110-5 COLD PUMP START-UP
HYTTHA INPUT DATA (CONTINUED)

27 9							
10.375	1.	.0580	4.	.0069	0.	6.	6.
28 9							
15.625	1.	.0580	4.	.0069	0.	6.	6.
29 9							
1.625	1.	.0580	4.	.0069	0.	6.	6.
30 9							
15.125	1.	.0580	4.	.0069	0.	6.	6.
31 9	0 0	1 135					
37.	1.	.0580	4.	.0069	0.	6.	6.
32 9	0 0	1 135					
6.625	1.	.0580	4.	.0069	0.	6.	6.
33 9							
15.25	1.	.0580	4.	.0069	0.	6.	6.
34 9							
15.125	1.	.0580	4.	.0069	0.	6.	6.
35 9							
10.	1.	.0580	4.	.0069	0.	6.	6.
36 9							
15.125	.375	.0220	4.	.0069	0.	6.	6.
37 9							
1.625	.375	.0220	4.	.0069	0.	6.	6.
38 9							
4.875	1.2	0.1	4.	.0069	0.	6.	6.
1 51	6 35	-38 -25	1	2 19	8		
9.	9.	5.716	8.155	8.529	15.	10.	175.
8.	3.	.63	5675.	6.	27.	69.	.0069
.69	.189	0.	0.	0.	0.	220.	4600.
4000.	3050.	2850.	25.	300.	8.5	50.	8.
0.0	7.0	8.0	13.0	17.0	20.0	32.0	200.0
0.	0.	500.	2000.	3000.	3900.	4000.	4025.
2 41	2 29	-30					
9.	1.661	1.	5.25	15.	1.0E-10	0.	0.
0.	0.	6.	.65	.0513			
3 11	2 2	-3 -6					
9.	.2	.69	2.3	5.9	5.1	.0069	0.
0.	0.	0.					
4 21	5 3	-4 2					
9.	9.	2.	.1	3.	.3	2.5	15.125
6.6	.033	.0069	1.	0.	0.	0.	0.
.022	.65						
0.0	200.						
0.0	0.0						
5 21	5 6	-7 3					
9.	9.	2.145	.1	3.	.3	2.5	12.
6.6	.033	.0069	1.	0.	0.	0.	0.
.022	.65						
0.0	100.	200.					
5.0	5.0	5.0					
6 69	3 4	-5					
9.		95.	45.	510.	1.	41.	400.
4000.	900.	4896.	.0069	.069	.075	2.91	0.
0.	0.	0.	0.	0.	.4		5.
7 11	2 26	7 -8					
9.	.2	.69	.9592	5.9	5.1	.0069	0.

TABLE 27 TEST RUN NO. 110-5 COLD PUMP START-UP
HYTTHA INPUT DATA (CONTINUED)

	0.		0.		0.					
8	11	2	34	-10	-9					
	9.		.2		.69	1.578	5.9	5.1	.0069	0.
	0.		0.		0.					
9	11	2	9	-12	-13					
	9.		.2		.69	2.3	5.9	5.1	.0069	0.
	0.		0.		0.					
10	11	2	10	-11	-14					
	9.		.2		.69	2.3	5.9	5.1	.0069	0.
	0.		0.		0.					
11	11	2	32	15	-33					
	9.		.2		.69	2.3	5.9	5.1	.0069	0.
	0.		0.		0.					
12	81	2	12	-19						
	9.		4.627		10.	.083	9.63	99.	69.	.0069
	1.		0.		0.	0.	0.	.01	.0002	
13	81	2	13	-18						
	9.		4.979		10.	.083	9.63	99.	69.	.0069
	1.		0.		0.	0.	0.	.01	.0002	
14	81	2	14	-17						
	9.		4.979		10.	.083	9.63	99.	69.	.0069
	1.		0.		0.	0.	0.	.01	.0002	
15	81	2	11	16						
	9.		4.627		10.	.083	9.63	99.	69.	.0069
	1.				0.	0.	0.	.01	.0002	
16	11			18	-20					
	9.		.2		.69	2.3	5.9	5.1	.0069	0.
	0.		0.		0.					
17	11	2	20	21	-22					
	9.		.2		.69	1.578	5.9	5.1	.0069	0.
	0.		0.		0.					
18	11	2	16	17	-21					
	9.		.2		.69	2.3	5.9	5.1	.0069	0.
	0.		0.		0.					
19	11	2	22	23	-24					
	9.		.2		.69	1.052	5.9	5.1	.0069	0.
	0.		0.		0.					
20	61	1	-27							
	3600.		0.		0.					
21	62	3	27	-23						
	9.		9.		17.36	93.410	1.558	4.67	5.467	1.75
	10.992		.11		665.	155.	.0069	.06	3.5	10.
	1.		0.		0.	0.	0.			
22	21	5	5	-26	2					
	9.		9.		2.	.1	3.	.3	2.5	12.
	6.6		.033		.0069	1.	0.	0.	0.	0.
	.022		.65							
	0.0		200.							
	0.0		0.0							
23	11	2	38	-28						
	9.		0.001		0.001	0.001	0.001	0.001	.0069	0.
	0.		0.		0.					
24	11	2	28	-1						
	4.		0.32146		0.61375	.8646	14.429	2.7772	.0069	0.
	0.		0.		0.					

TABLE 27 TEST RUN NO. 110-5 COLD PUMP START-UP
HYTTHA INPUT DATA (CONTINUED)

```
25   11    2    1  -29
     4.   0.32146   0.61375        1.    14.429   2.7772    .0069    0.
     0.        0.         0.
26   11   .2   30   -2
     4.   0.32146   0.61375        .5    14.429   2.7772    .0069    0.
     0.        0.         0.
27   11    2    8  -31
     4.     .5214   0.61375        1.    14.429   2.7772    .0069    0.
     0.        0.         0.
28   11    2   31  -32
     4.     1.217   0.61375        1.    14.429   2.7772    .0069    0.
     0.        0.         0.
29   11    2   33  -34
     4.   0.32146   0.61375        1.    14.429   2.7772    .0069    0.
     0.        0.         0.
30   11    2   24  -35
     4.   0.32146   0.61375        1.    14.429   2.7772    .0069    0.
     0.        0.         0.
31   11    2   25  -36
     4.   0.32146   0.61375     .5742    14.429   2.7772    .0069    0.
     0.        0.         0.
32   11    2   36  -37
     4.   0.32146   0.61375     .3080    14.429   2.7772    .0069    0.
     0.        0.         0.
33   11    2   37  -15
     9.     .0726       .69        1.       5.9      5.1    .0069    0.
     0.        0.         0.
15   19    1
 1   12    1    5         16.
19    3    0   24   30      1    0   35    1    1
 2    1    2    8         12.
 1    2    0   38   23      1    0   28   24    1    0    1   25    1    0   29
 3    2    3    5         12.
 2    1    0   30   26      1    0    2    3    1
 4    3    4    5         12.
 3    3    0    6    5      1    0    7    7    2
 5    3    4    9          0.
 3    2    0    3    4      1    0    4    6    1    0    5   22    1    0   26
 7    1
 6    4    5    7         12.
 7    3    0    8   27      1    0   31   28    1    0   32   11    1
 7    5    6    5         16.
11    3    0   33   29      1    0   34    8    1
 8    6    7    3          8.
 8    3    0    9    9      1
 9    6    8    3          8.
 8    2    0   10   10      1
10    7   10    5          4.
 9    2    0   12   12      1    0   19   16    1
11    7   10    5          4.
 9    3    0   13   13      1    0   18   16    2
12    8   11    5          4.
10    3    0   14   14      1    0   17   18    2
13    8   11    5          4.
10    2    0   11   15      1    0   16   18    1
14   10    9    3          8.
16    3    0   20   17      1
15   11    9    3          8.
18    3    0   21   17      2
16    9   12    3         16.
17    3    0   22   19      1
17   13   12    3         .01
21    2    0   23   19      2
18   15   14    3         .01    3000.      3000.
20    1    0   27   21      1
19    1    5    9          4.
 1    3    0   25   31      1    0   36   32    1    0   37   33    1    0   15
11    2
 7    3    0   00    0
 1    4    1    4   -1     -4
 2    2    1    4
23    2   -2   -4
31    3    1    4   -2
33    2   -1   -4
35    1    4
36    3    1    4   -2
 2    4   21    4   21      5
```

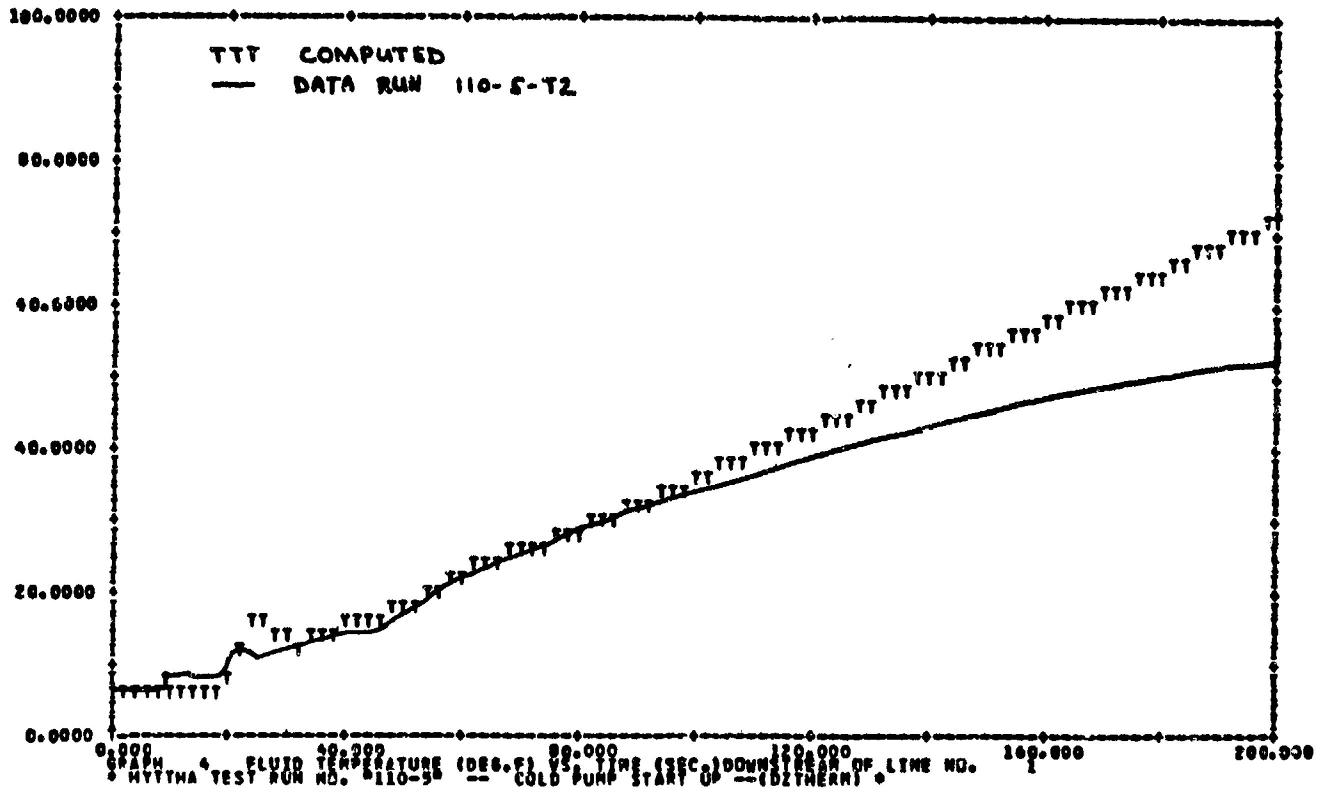


FIGURE 176 HYTTHA SIMULATION DATA RUN 110-5-T2

SECTION VI

HYDRAULIC LINE MECHANICAL RESPONSE (HLMR) PROGRAM

1. BACKGROUND

Hydraulic line vibrations due to pulsations from axial piston-type pumps can cause wear and tear problems in high performance aircraft. In a continuing effort, analytical tools are being developed to correlate theory and test data. Complicating the analysis is the fact that piping systems contain bends which tend to change the mechanical response. This can cause fundamental resonances in the pump operating regime and damage the installation.

The importance of developing analytical tools coupled with experimental tests was recognized at AFAPL and funds were allocated to pursue this development.

The initial objectives of the HLMR effort were to:

- o Develop and verify a computer program for predicting line mechanical response due to pump pulsations
- o Provide basic data for the design of piping systems
- o Provide information regarding intermediate supports

The test program involved the determination of mode shapes of mechanical and hydraulic resonances due to pump excitation as measured by accelerometers on three configurations. These were a straight pipe, a pipe with a single 90-deg bend, and one with two 90-deg bends (dogleg). In addition, the effect of an intermediate elastomeric support was evaluated.

The results were reported in reference (2). There was good correlation between analyses and test data using simplifying assumptions.

Since completion of these first series of test, supplemental funds were allocated to repeat the test on the specimen with a single 90-deg bend using strain gages in lieu of accelerometers.

To review the first series of tests performed on the one-elbow pipe (one-inch titanium line with 0.051-in. wall thickness) the results are summarized in Figure 177. The horizontal lines indicate natural frequencies measured in the test. The radial lines depict the relationship between the exciting frequency and the pump speed, representing the harmonics or multiples of the pumping frequency. The intersections of the radial lines (hydraulic exciting frequencies) with the horizontal

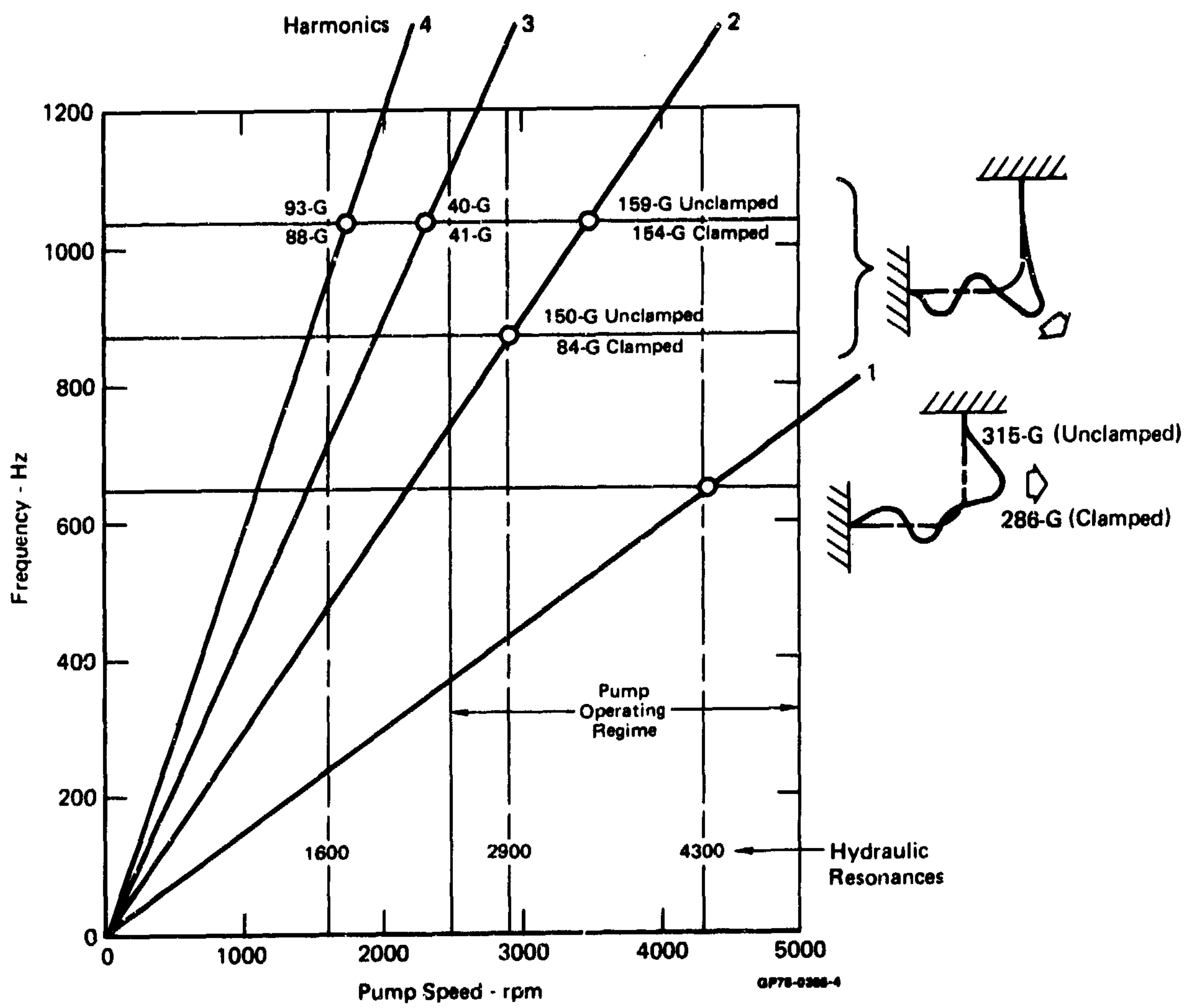


FIGURE 177 HLMR TEST DATA SUMMARY
One-Elbow Pipe

lines (natural mechanical frequencies) indicate conditions of mechanical resonance.

Although the tests indicated three hydraulic resonances, at pump speeds of 1600 rpm, 2900 rpm, and 4300 rpm (predicted by the HSFR program) none excited a fundamental mechanical resonance. At 1600 rpm, no significant mechanical responses were measured in any of the test configurations. At the other two pump speeds there were larger responses in terms of measured accelerations. By extrapolating the data, the intersection of the horizontal 1030 Hz line with the radial first harmonic line, indicates that the fundamental mechanical resonance would occur at 6900 rpm. Fortunately the pump does not have the capability to achieve such a speed where a potential catastrophe would most likely occur.

The mode shapes are shown as insets in Figure 177. The elastomer in the clamp was elastic enough that there was no effect on the mode shape and the accelerations were only slightly lower.

2. ANALYTICAL APPROACHES

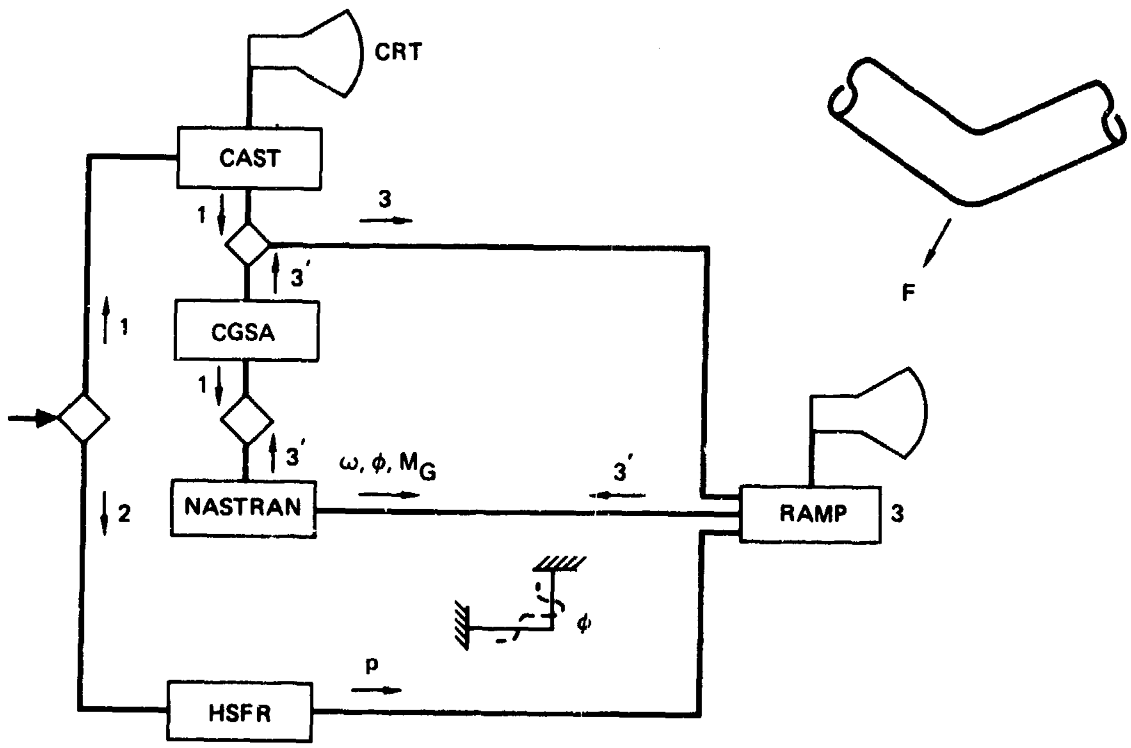
The most important aspect in developing an HLMR program is selection of a mathematical model. The best mathematical solution will not overcome a modeling error.

The natural frequencies/critical speeds are unique to a given line installation. The equations of motion can be derived without too much difficulty. The resolution of these equations is the basic obstacle. The stumbling block is either the geometric arrangement or the irregular shape of the system. This is where rigorous, exact closed form, mathematical solutions are inadequate.

a) Technical Overview

A preliminary study was made to investigate methods for calculating hydraulic line vibration response and strains. Vibration properties could be computed in NASTRAN and would serve as the base for these response calculations which would be performed outside of NASTRAN using MCAIR based programs. A brief discussion is given here with more details in Appendix

Figure 178 shows a flow chart of the proposed computer program latch-up where NASTRAN is interfaced with MCAIR's programs; Computer Aided Structural Technology (CAST), Computer Graphics Structural Analysis (CGSA), Random Access Matrix Program (RAMP), and HSFR. Computer graphics CRT set-up would be used as much as possible to enhance visual displays, to accurately check and monitor work and to increase understanding.



Notes:

1. CAST/CGSA/NASTRAN/RAMP
   Model construction, calculation of frequency (ω), mode shape (φ) and generalized mass, pass to RAMP
2. HSFR
   Calculate pressure (p), pass to RAMP
3. RAMP
   Calculate forcing function, F, and response, pass to CAST for display
3. HSFR/RAMP/NASTRAN/CAST
   Pressure data to RAMP calculation of forcing function, then to NASTRAN for response, display in CAST

GP83-0077-108

FIGURE 178 - FLOWCHART OF LINE VIBRATION RESPONSE PROGRAM

A NASTRAN model would be constructed using MCAIR's CGSA procedure. The model would be kept simple, but representative enough to allow for proper definition of radial,in-plane and out-of-plane dynamics. Various clamp locations along the pipe will be allowed, and various types of clamps and stiffness ranges will be investigated. The vibratory data derived from NASTRAN will consist of normal mode frequencies, shapes, and generalized masses. These data will be passed to RAMP where the response functions and strains will be computed using the pump pressure functions from HSFR.

Consideration was given to including to the extent possible both (a) the response due to steady state vibration forces induced by the pressure pulsations (combination of fluidic and mechanical resonances), and (b) the response due to the Bourdon-Tube type loads in pipe bends. This latter forcing function was found to occur in the curved part of the pipe, and will couple the various motions in the radial in-plane and out-of-plane directions. This force can also lead to unstable, self-excited, vibratory conditions in some cases. This latter mechanism is similar to that found in "follower-force" problems and in flutter, see References (13) and (14). The plan is to primarily concentrate on this Bourdon-Force excitation which may be the key to large responses noted, whether steady state vibratory excitation or self-excited cases. A preliminary check was made on a self-excited vibration case using approximate data. This showed the possibility for oscillations to occur and involve the Bourdon Force coupling of the first and third out-of-plane bending modes. This mechanism was not observed in the tests because we were unable to excite the first mode frequency in the range of pump operating frequencies.

A limited effort would be given to evaluating nonlinear effects and other oscillatory effects such as those which might arise, due to oscillatory forces in the Mathieu-Hill mechanism.

b) Equations of motion

The equations of motion using normal vibration modes appear as:

$$[M_{jj}]\ \{\ddot{q}_j\} + [2\ M_{jj}\ \zeta_j\ \omega_j]\ \{\dot{q}_j\} + [\omega_j^{\ 2}\ M_{jj}]\ \{q_j\} = \{F\} \tag{63}$$

where M_{jj} is generalized mass, ω_j is normal mode frequency, ζ_j is normal mode damping coefficient, q_i is the normal mode degree of freedom deflection amplitude, and F_j is the generalized force. The generalized force can be broken down into

$$F_j = F_{o_j} + F_{c_j} \tag{64}$$

where F_{oj} is a generalized force independent of the deflection q_i, while F_{cj} is the generalized force varying with the deflection. Each are ultimately functions of time.

Cases involving forces independent of the deflection, F_o, give rise only to steady state response and are stable. However, cases involving forces that are directly proportional to the deflection, F_c, can give rise to instability or self-excited vibrations as found in "follower force" problems, see Ref.(13), and in flutter, see Ref.(14).

where F_{o_j} is a generalized force varying only with time, but independent of the amplitude Z, while F_{c_j} is the generalized force varying with the amplitude Z and is thus also a time function (indirectly).

Cases involving forces independent of the coordinates, F_o, give rise only to steady state response, and are as stable as the input force. However, cases involving forces that are directly proportional to the coordinates, F_c, can give rise to instability or self-excited vibrations as found in "follower force" problems, see Ref. (13) and in flutter, see Ref. (14).

We can represent F_{c_j} as

$$\{F_{c_j}\} = [A_{jk}]\ \{\ddot{q}_k\} + [B]\ \{\dot{q}_k\} + [C]\ \{q_k\} \tag{65}$$

and substituting into (1) yields

$$\begin{aligned} &\Big[[M_{jj}] - [A_{jk}]\Big]\ \{\ddot{q}_k\} + \Big[[2\ M_{jj}\ \zeta_j\ \omega_j] - [B_{jk}]\Big]\ \{\dot{q}_k\} + \\ &\Big[[\omega_j^{\ 2}\ M_{jj}] + [C_{jk}]\Big]\ \{q_k\} = \{F_{o_j}\} \end{aligned} \tag{66}$$

Equation (66) gives all possible solutions for stability and response. The left hand side with $F_o = 0$ gives the transient solution, say q_T, and hence stability.

The solution with the right hand side not zero, $F_o \neq 0$, gives the steady state response solution, q_S. The total solution is the sum

$$q = q_T + q_S \tag{67}$$

The actual physical deflections, Z, are a product of the normal mode shapes Φ, from NASTRAN in this case, and the q's or,

$$Z = \Phi q = \Phi(q_T + q_S) \tag{68}$$

The structural strains are computed from the curvature of Z with respect to distance along the pipe, or for example,

$$\epsilon_X \sim u \frac{\partial^2 Z}{\partial X^2} = u \frac{\partial^2 \phi}{\partial X^2} (q_T + q_S) \tag{69}$$

where u is the distance from a mid plane to the extent where maximum strain occurs.

Returning to Equation (66) we can indicate some preliminary results of a study of the self excited vibration case. For our approach, the Bourdon-Tube Load (L_B) is idealized as indicated on Figure 179, and has a magnitude given by

$$d(L_B) = \quad pr^2 \, d\phi \tag{70}$$

where p is the pressure, r is the pipe radius, $d\phi$ is in differential angle along the pipe bend. Based on the idea for vertical excitation due to (L_B), or (L_{B_V}) where

$$d(L_{B_V}) = d(L_B) \frac{\partial Z}{\partial n} = d(L_B) \left[\frac{\partial Z}{\partial X}\frac{\partial X}{\partial n} + \frac{\partial Z}{\partial Y}\frac{\partial Y}{\partial n}\right] \tag{71}$$

note

$$\left.\begin{aligned} X &= n \cos \phi \\ Y &= n \sin \phi \end{aligned}\right\} \tag{72}$$

along the curved section $0 < \phi < 90°$,

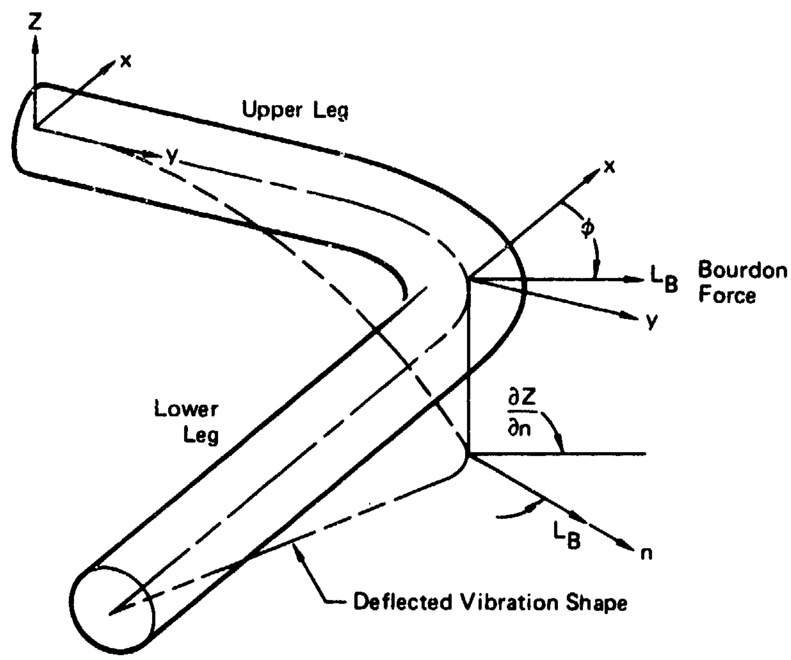
$$d(L_B)_V = d(L_B) \left[(\cos \phi) \frac{\partial Z}{\partial X} + (\sin \phi) \frac{\partial Z}{\partial y}\right] \tag{73}$$

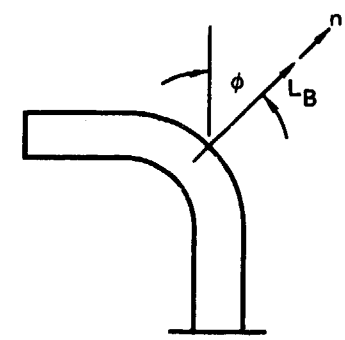
The virtual work of this force is δW, where

$$\delta(dW) = d(L_B)_V \, \delta Z \tag{74}$$

and the generalized force F is defined along the curved part of the line as

$$F_j = \int_0^{\pi/2} \frac{\delta(dW)}{\delta q_j} = \int_0^{\pi/2} \frac{(\delta Z)}{\delta q_j} (dL_B)_V \tag{75}$$

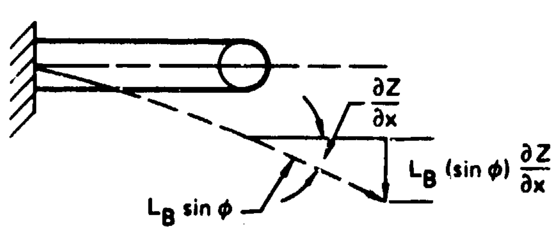




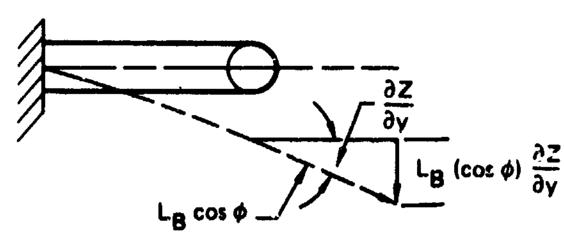
n cos φ = x
n sin φ = y

$$\frac{\partial Z}{\partial n} = \frac{\partial Z}{\partial x}\frac{\partial x}{\partial n} + \frac{\partial Z}{\partial y}\frac{\partial y}{\partial n} = \frac{\partial Z}{\partial x}(\cos\phi) + \frac{\partial Z}{\partial y}(\sin\phi)$$

Side View - Upper Leg



Side View - Lower Leg



$$(L_B)_V = L_B\frac{\partial Z}{\partial n} = L_B(\cos\phi)\frac{\partial Z}{\partial x} + (\sin\phi)\frac{\partial Z}{\partial y}$$

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FIGURE 179 - BOURDON TUBE LOADS

Note that the deflections Z defined in Equation (68) can be expressed as

$$Z = \sum_{J=1}^{N} \Phi_J \, q_J, \quad Z = \sum_{K=1}^{N} \Phi_K \, q_K \tag{76}$$

and substituting this into Equation (75)

$$F_j = \int_0^{\pi/2} (\Phi_J) \, d(L_B) \quad . \tag{77}$$

but again using Equations (70), (73) and (76) into (77), and the relationship

$$\frac{\partial \Phi}{\partial X} = \frac{\partial \Phi}{\partial S} \cdot \frac{\partial S}{\partial X} \tag{78a}$$

$$\frac{\partial \Phi}{\partial Y} = \frac{\partial \Phi}{\partial S} \frac{\partial S}{\partial Y} \tag{78b}$$

the forces F_J can be expressed as

$$F_J = \left[4pr^2 \int_0^{\pi/2} \Sigma \, \frac{\partial \Phi_K}{\partial S} \{(\cos \phi) \frac{\partial S}{\partial X} + (\sin \phi) \frac{\partial S}{\partial Y}\} \, \Phi_J \, d\varphi \right] q_K \tag{79}$$

Thus, in this case, the terms [A] = [B] = 0 in Equation (66) while the $[C_{JK}]$ term is expressing $d\phi = R^{-1}dS$

$$C_{JK} = \frac{\partial F_J}{\partial q_K} = 4pr^2 \int_0^{\pi/2} \left(\frac{\partial \Phi_K}{\partial S}\right) \left[(\cos \phi) \frac{\partial S}{\partial X} + (\sin \phi) \frac{\partial S}{\partial Y}\right] \Phi_J \quad R^{-1} ds \tag{80}$$

Once the mode shapes Φ are known, the loads can be determined.

For stability assessment, a search of the roots of the determinant is made of the left hand side of Equation 66. For example, for a multidegree of freedom case, the determinant using the Bourdon Load shown here is

$$\mathrm{DET} \left| [\omega^2 M_{JJ}] + [2\zeta_J M_{JJ} \, \omega_J \, \omega] + [[\omega_J^2 M_{JJ}] - p[D]] \right| = 0 \tag{81}$$

where $p[D] = [C]$.

The roots ω will be a polynomial dependent upon pressure (p), or

$$\sum_{n=1}^{r} A_n [\omega(p)]^n = 0 \tag{82}$$

This can have real and or complex roots indicating stable, neutrally stable, or unstable behavior. This is shown by the expression

$$q_T = \sum_{n=1}^{2J} C_N e^{i\omega_N t} \tag{83}$$

where the C_N terms are constants associated with initial conditions, and the ω_N terms are the roots of Equation (81). If the ω's are real below the critical pressure, stable motion exists. Above the critical pressure some roots have complex form indicating unstable motion.

An exploratory case was carried out, using a two degree of freedom configuration considering the 1st and 3rd out-of-plane pipe bending modes. This two degree of freedom case has a closed form solution for critical pressure and frequency. Lacking precise mode shape data it was determined that the frequencies coalesce as depicted in Figure 180.

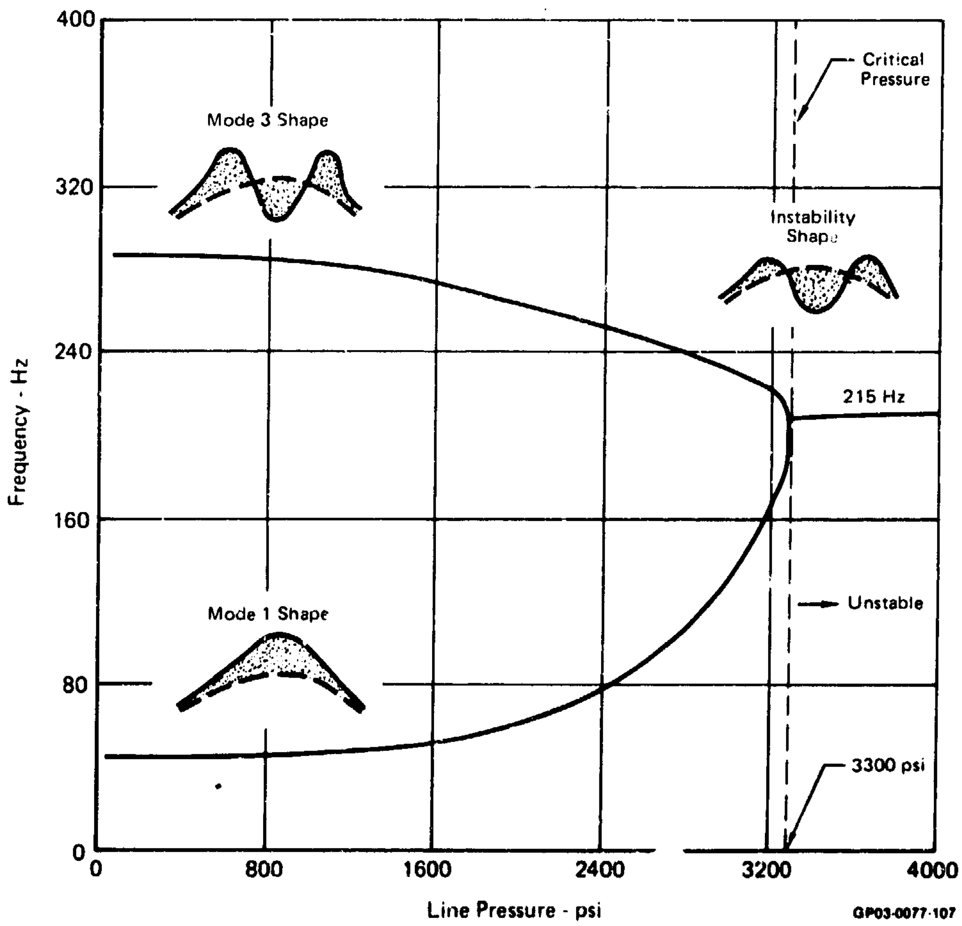


FIGURE 180 · FREQUENCY vs LINE PRESSURE
2 D/F Case Modes 1 and 3

3. VERIFICATION TESTS

Proof of theoretical analyses is in the prediction of significant parameters affecting a given hardware system. This means that the accumulation of test data affords analysis a certain degree of confidence in the ability to extrapolate the data.

a. Test Set - Up

Previous test response data was taken with accelerometers. Subsequent tests were performed with one of the original test specimens, the one-elbow pipe, using strain gages to measure response. Since pulsation response is significant, the primary objective in this series of tests was to evaluate the effect of stresses on the hydraulic line fatigue life, and secondarily to reconfirm the hydraulic/ mechanical resonances.

The test specimen was one-inch diameter tubes, with 0.051-in. wall thickness of 3Al-2.5V titanium having a single 90-deg bend at midpoint. The pipe was rigidly mounted at each end using Dynatube fittings between the brackets, Figure 181, attached to a steel plate. In the test set-up, the test specimen was a part of the pressure system between the pump and the flow control valve, and was isolated from extraneous mechanical vibration inputs in order to more clearly evaluate the effect of the pump pulsations. The test fluid was MIL-H-5606C hydraulic oil.

Also shown in Figure 181 are the locations of the strain gages, four gages at each of the five selected locations, and the position of the clamp block. The type of gages installed were the CEA-06-125UW-350 with a gage factor of 2.09 $\pm$0.5%. A gage factor is defined as the unit change of resistance divided by the strain which is a function of the gage material.

At each location the gages are attached to the tubing 90-deg apart; one each on top and bottom, one each on the inside and outside of the plane containing the bend.

A typical strain gage installation is shown in Figure 182. The strain gages were connected to the lab instrumentation through signal conditioning equipment by means of a five-wire system and shunt calibration. Only ten strain gage outputs could be recorded on analog tape at a given time. Consequently two runs were made for a given test condition to record the in-plane and out-of-plane values.

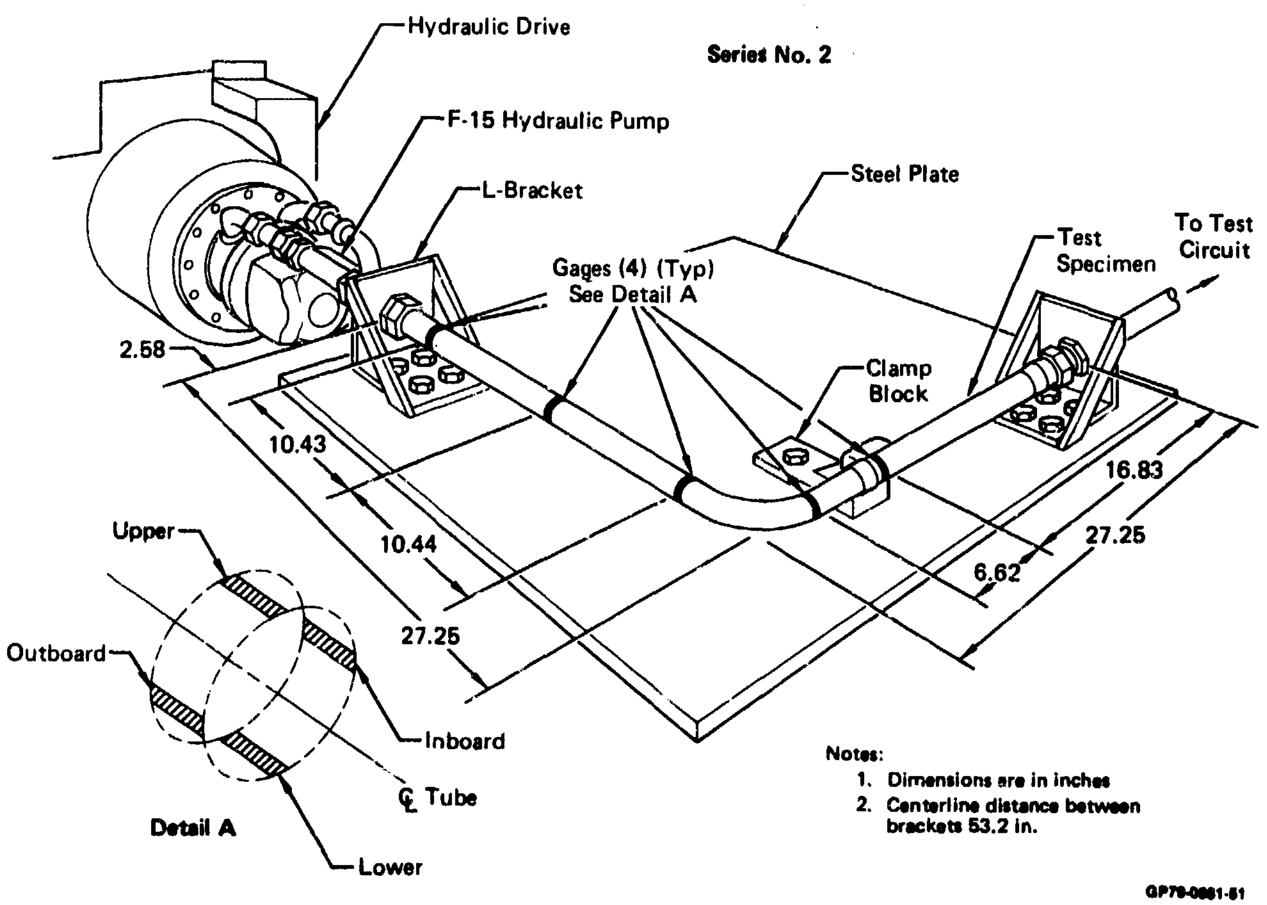


FIGURE 181 HLMR TEST SET-UP OVERVIEW

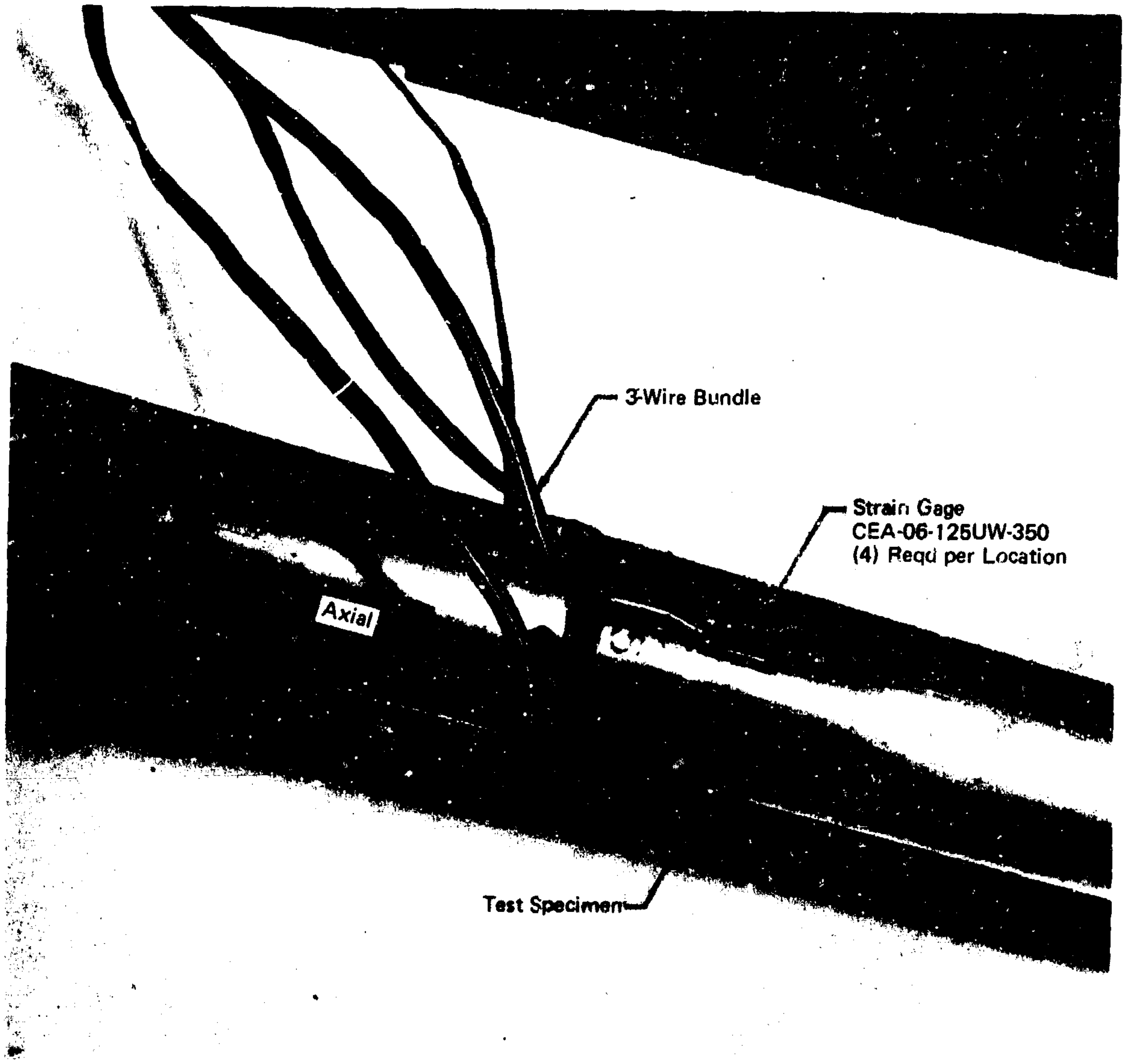


FIGURE 182 TYPICAL STRAIN GAGE SETUP

All tests were performed with a pump outlet flow of 2.0 gpm, at a pump inlet temperature of 130 $\pm$ 10°F, and a system air content of less than one percent by volume.

A transducer, coupler, and a clamp-on block were used to determine the standing pressure wave in the test specimen. The procedure was to place the clamp-on transducer at different locations on the test specimen and conducting 1000 rpm to 5000 rpm pump sweeps while recording the fundamental pressure amplitude. The standing pressure waves were determined for the unclamped and clamped conditions.

From the analog tape data, different harmonic strain amplitudes as a function of pump speed were made for each strain gage using the same technique as the one for recording the standing pressure wave.

For test Series No. 1 the clamp used was a single loop clamp, Figure 183, with a yellow nitrile elastomer cushion. The results obtained were similar to those with the specimen unclamped, as previously mentioned in Part I of Section VI. (Ref. 2) For this second set of tests, it was decided to use an F-15-type clamp, also shown in Figure 183, which uses a white teflon grommet as a spacer in conjunction with a single loop clamp with an .02-in. thick green teflon cover. The installed clamp is shown in Figure 184 and was designed to restrain the test specimen in the inplane direction since the first test series indicated that this was the direction of maximum responses.

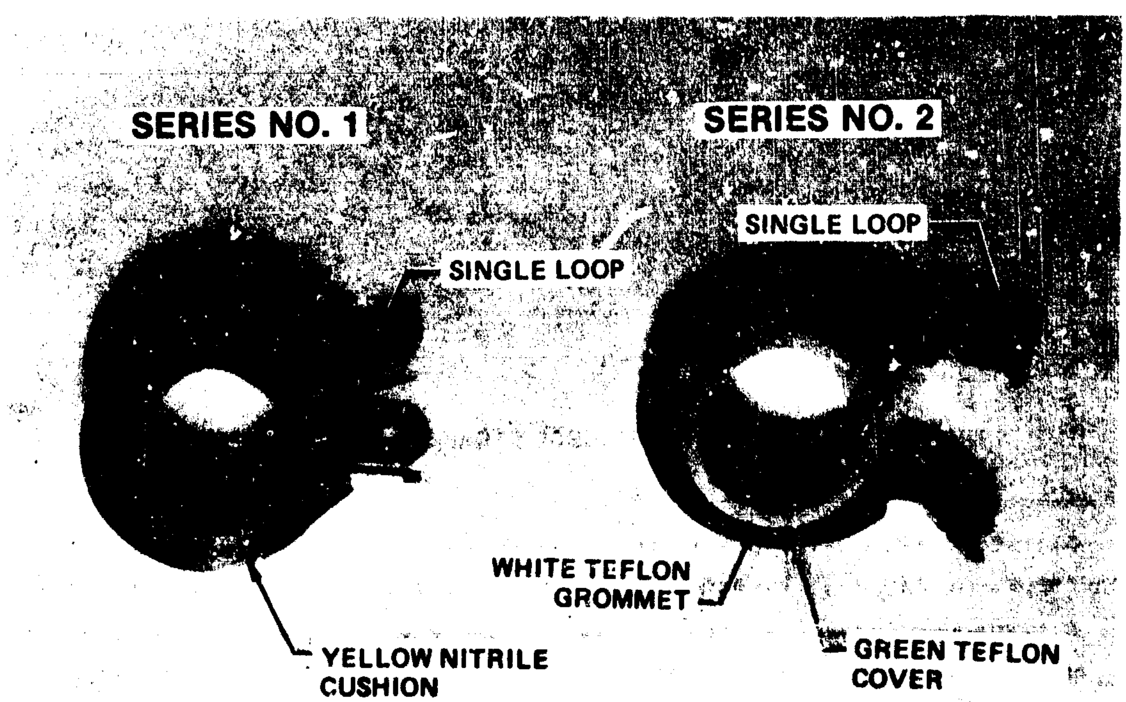
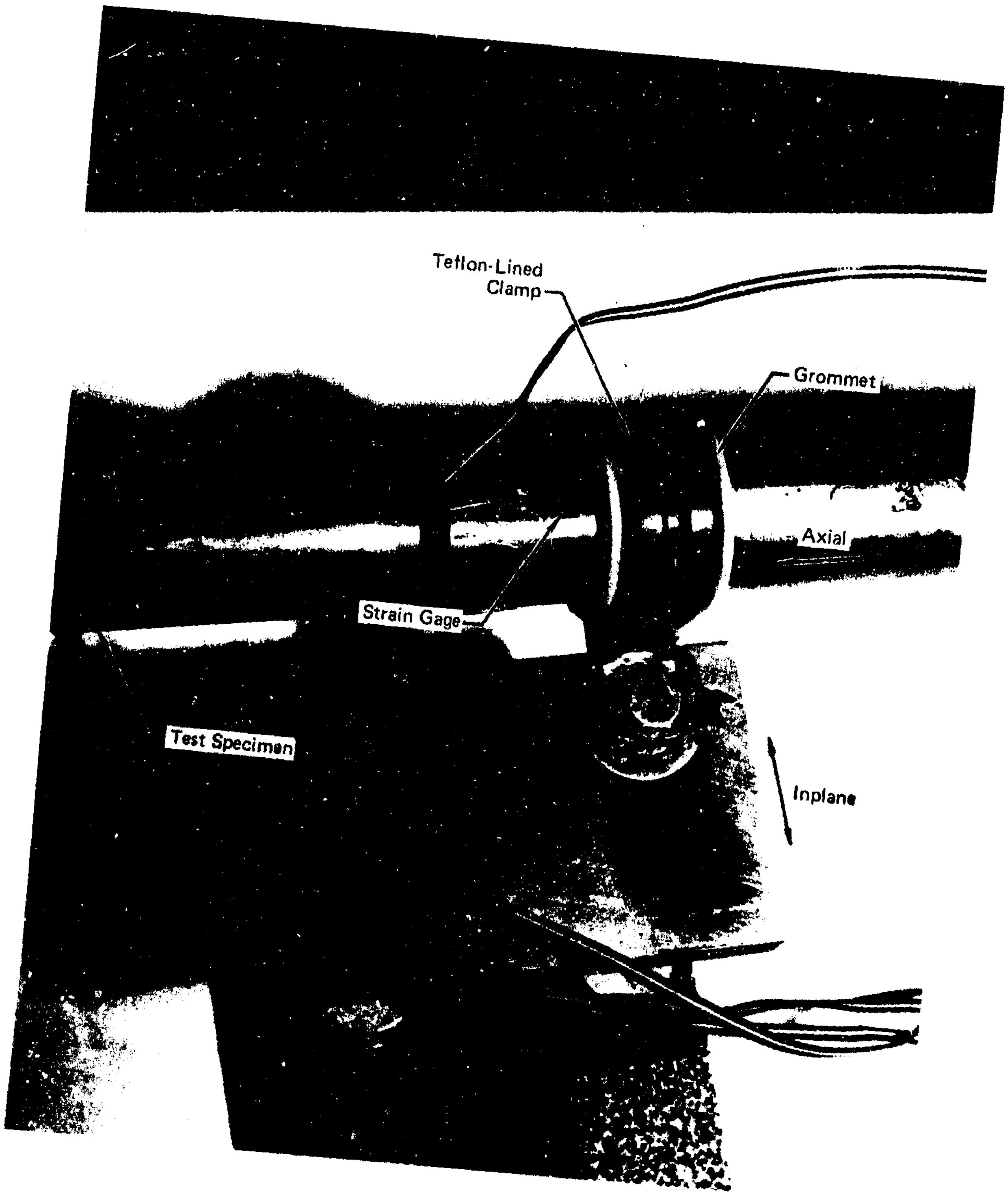


FIGURE 183 TEST CLAMPS



GP79-0881-47

FIGURE 184 CLAMP INSTALLATION

b. Test Results

The hydraulic resonances were the same three as those reported in Reference (2), namely, 1600 rpm, 2900 rpm, and 4300 rpm.

Examples of the pump sweeps to determine resonances are seen in Figures 185 through 11. In Figure 185 at the indicated location with the test specimen unclamped, the resonances are clearly discernible especially the one at 4300 rpm which is being excited by the major hydraulic resonance. At the same location with the test specimen clamped, Figure 186, again the hydraulic resonance at 4300 rpm produced a significant strain response and another one at 1350 rpm which is a mechanical response. However the strain measurements as a function of pump speed must be compared for all locations or an erroneous picture may be obtained as depicted in Figure 187. Here the location of the measurement is near the clamp and the responses are much more subdued at 1350 rpm and the major response is at 1600 rpm which has been previously identified as a hydraulic resonance.

Once the composite mechanical and hydraulic resonances were identified, the corresponding pump speeds were dwelled at for approximately one minute. Data plots of strains as a function of frequency at each resonant pump speed indicates the harmonic content. For example, at 3436 rpm (a mechanical resonance) and at the location indicated in Figure 188, the unclamped test specimen indicates that the second harmonic of the pump speed produces the largest response. Figures 189 and 190 are the strain responses of dwells at the hydraulic resonances of 2860 rpm and 4300 rpm, respectively. In Figure 189, again the second harmonic is the predominant response. However in Figure 190 the major hydraulic resonance at 4300 rpm does produce a first harmonic content.

The results are summarized in Figures 191 and 192 for the unclamped and clamped versions, respectively. Similar to the summary for the first test series, the horizontal lines are the measured natural frequencies and the radial lines are harmonics of the pump speed. The basic reference is that 5000 rpm represents 750 Hz. The resonances are the intersections of the radial lines with the horizontal lines. The unclamped test specimen duplicated the first test series and in addition produced a mechanical response at the 1600 rpm hydraulic resonance not previously noted. However, this is only of passing interest as the pump operating regime is considerably higher than this value. All the remaining mechanical and hydraulic resonances produced identical frequencies. The clamped test specimen, had the same hydraulic responses and frequencies, Figure 192, but the more rigid clamp arrangement caused the mechanical resonances to drop to the first harmonic radial line and consequently produced lower frequencies. This indicates, once again, that the use of clamp(s) in long lines can and does produce undesirable results and eventual wear of the teflon grommet and ultimately metal-to-metal fretting.

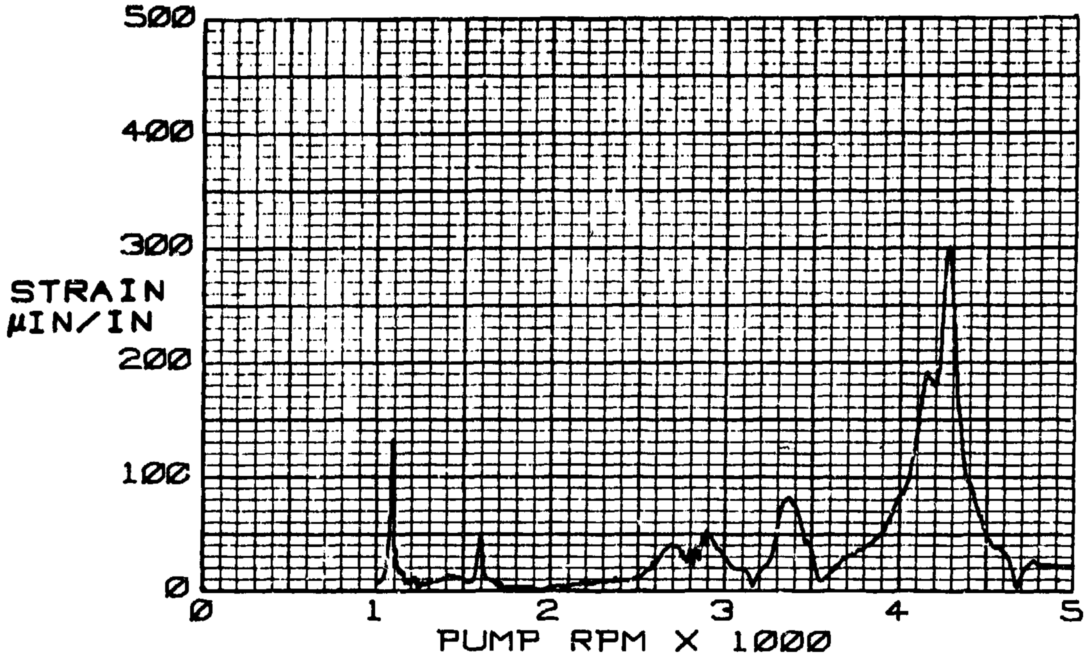


FIGURE 185 ONE-ELBOW UNCLAMPED,
111-Ø4-S4, FUNDAMENTAL,
LOCATION 1, OUTBOARD

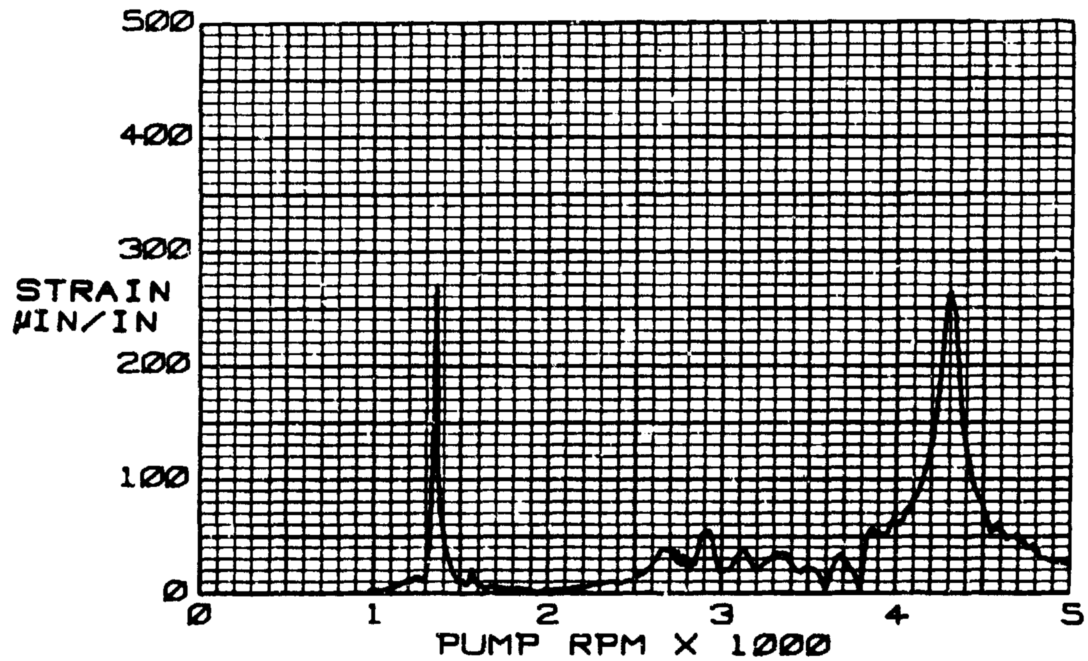


FIGURE 186 ONE-ELBOW CLAMPED,
111-Ø5-S4, FUNDAMENTAL,
LOCATION 1, OUTBOARD

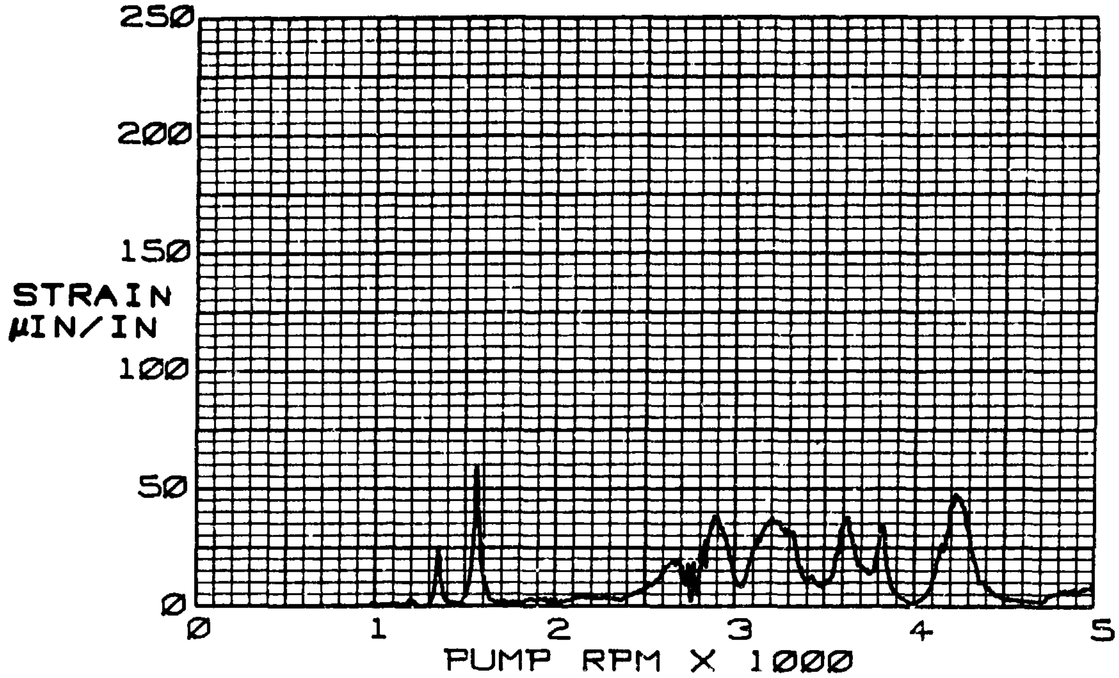


FIGURE 187 ONE-ELBOW CLAMPED,
111-Ø5-S15, FUNDAMENTAL,
LOCATION 4, LOWER

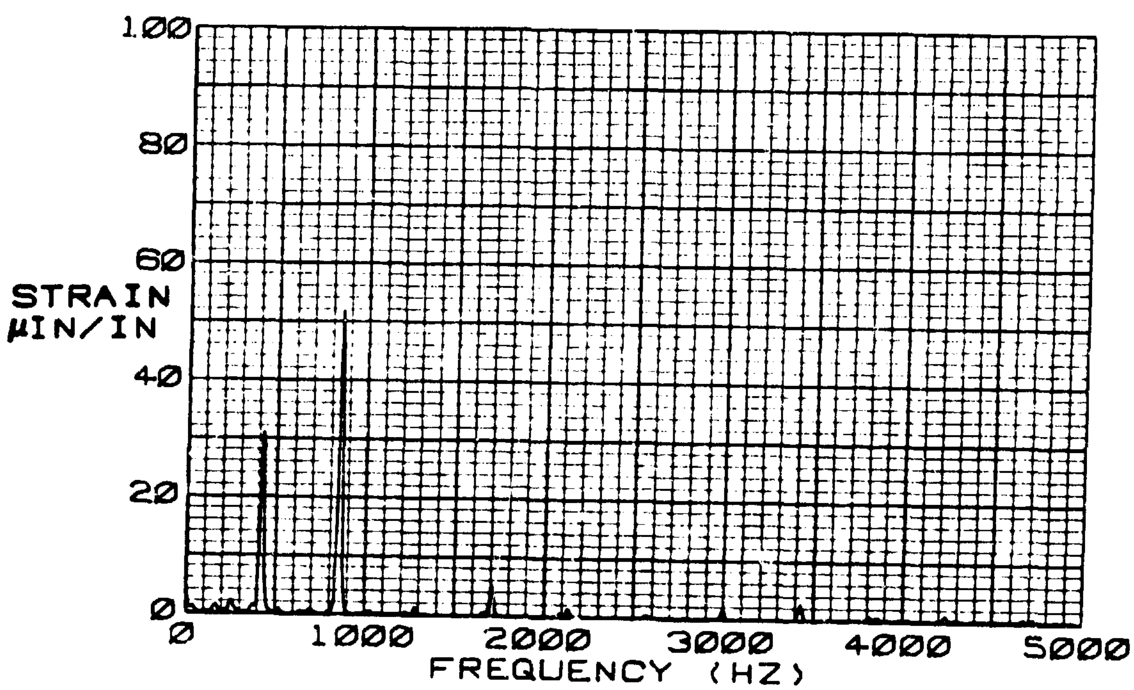


FIGURE 188 ONE-ELBOW PIPE UNCLAMPED,
111-17-S17, 286Ø RPM,
LOCATION 5, UPPER

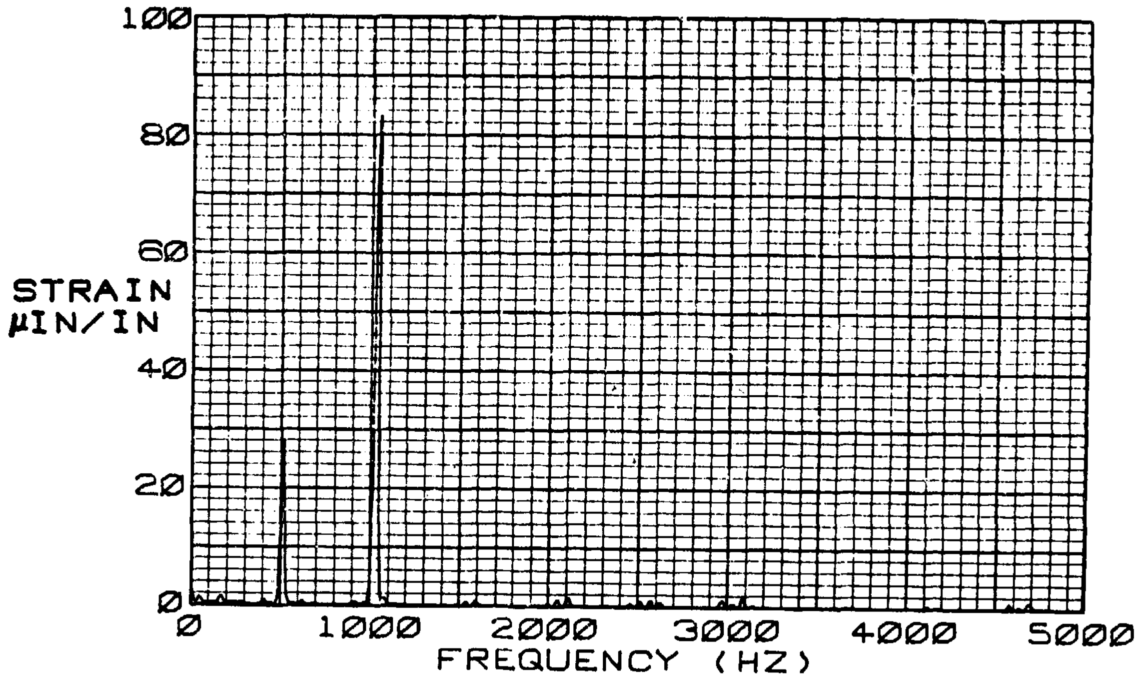


FIGURE 189 ONE-ELBOW PIPE UNCLAMPED,
111-18-S17, 3436 RPM,
LOCATION 5, UPPER

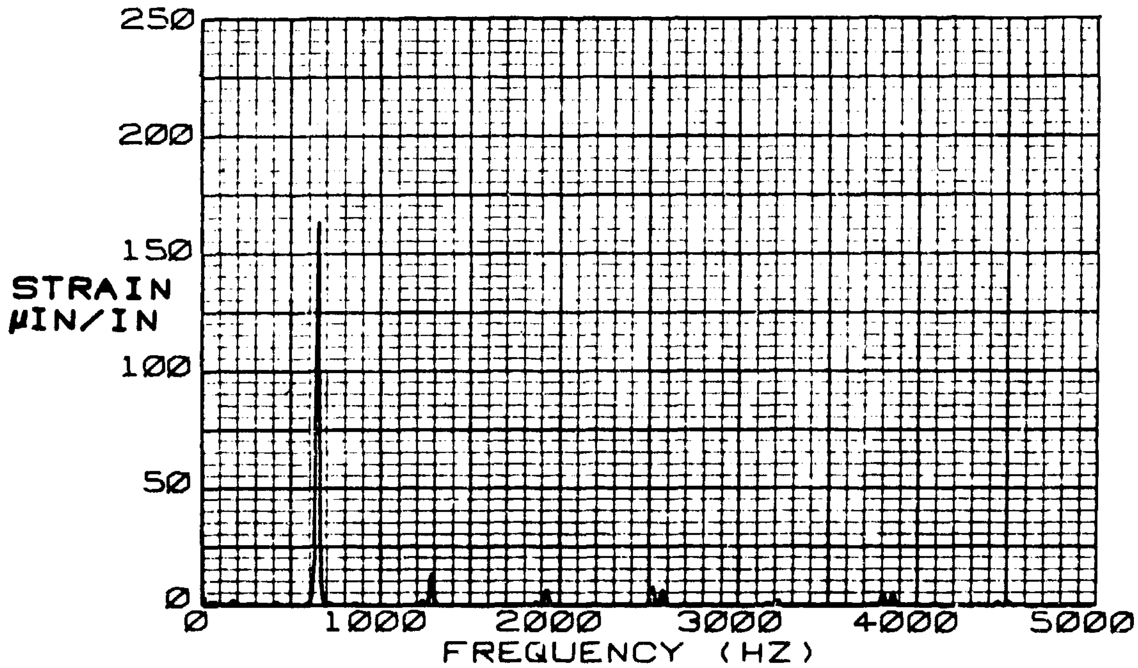


FIGURE 190 ONE-ELBOW PIPE UNCLAMPED,
111-19-S1, 4303 RPM,
LOCATION 1, UPPER

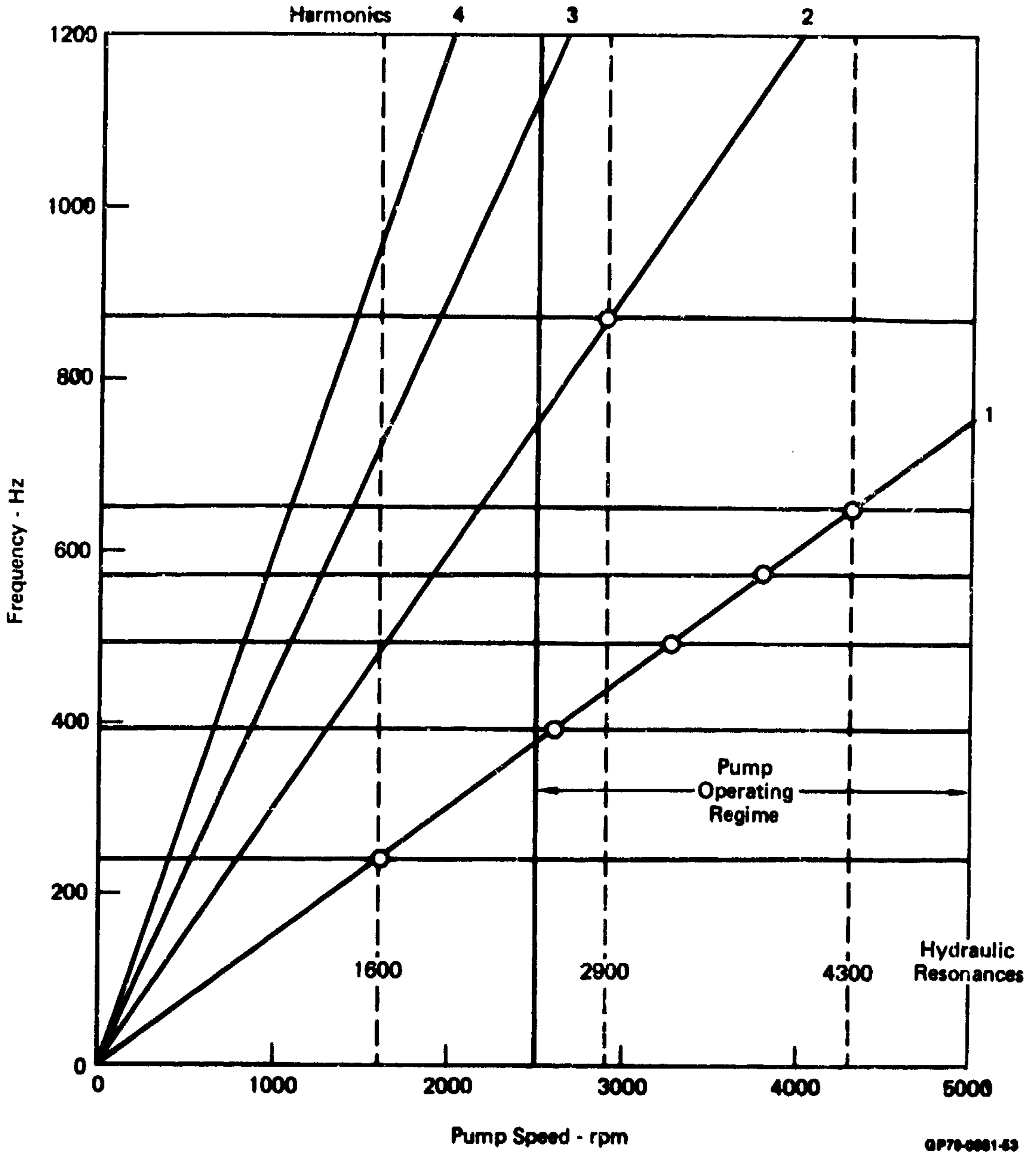


FIGURE 191 HLMR TEST DATA SUMMARY ONE-ELBOW PIPE UNCLAMPED
Based on Strain Gage Measurements

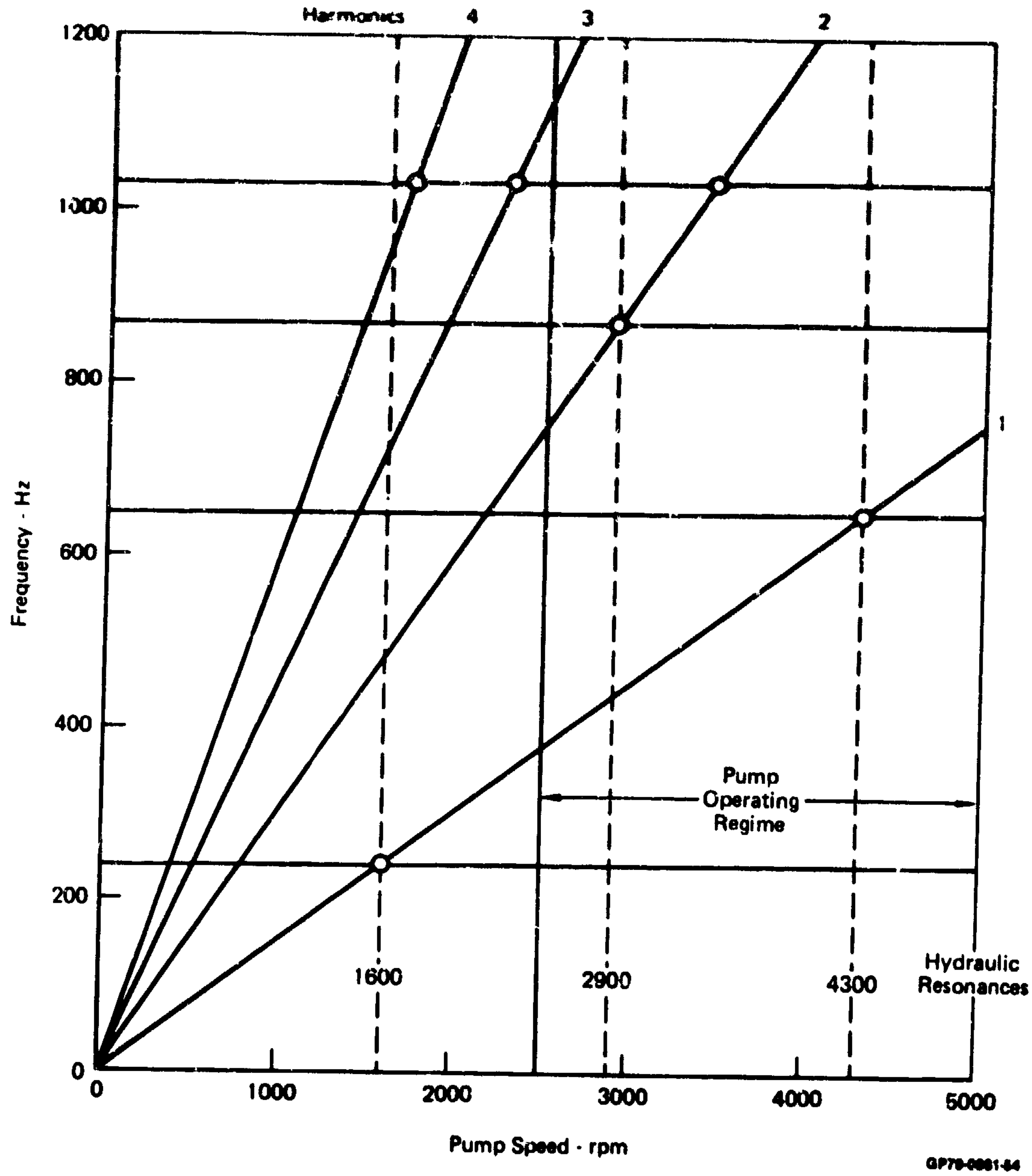


FIGURE 192 HLMR TEST DATA SUMMARY ONE-ELBOW PIPE CLAMPED
Based on Strain Gage Measurements

c. Data Reduction Procedure

The longitudinal strain measurements at each location would produce erroneous stress values if taken individually. Since radial or hoop strains have been consistenly measured at an average of 1220 micro in./in. independent of pump speed, the procedure was to use three longitudinal strain measurements and the hoop strains value at each location to determine the four associated parameters. These are the internal pressure, the axial load, and the inplane and out-of-plane moments. Only the internal pressure value is used to determine the loop stress which, in turn, provides a indication of the static strength of the specimen. But all four values go into the determination of the longitudinal or axial stress which is a measure of the specimen fatigue life. Consequently an assessment of the static strength and fatigue life of a given specimen is a measure of its structural integrity.

From the first series of tests, the six accelerometer data points for each resonance was used on a beam with built-in ends. This was performed only for the unclamped case. The method used converts the accelerometer loading successively through the shear and moment values, and from the moment the bending stresses as outlined in Appendix W.

d. Stress Comparison

From the above mentioned methods, the stresses for the unclamped specimen based on accelerometer and strain gage data are compared in Figures 193 through 197. These five graphs cover the mechanical and hydraulic resonances. The accelerometer data indicate the inplane stresses are higher than the out-of-plane stresses especially at the major hydraulic resonance at 4300 rpm. The calculated built-in stresses at the brackets are also shown.

The longitudinal stresses based on the strain gage measurements are the total values for each location and do not differentiate between the inplane and out-of-plane directions. Although these stresses are not directly comparable to the data derived from the accelerometer data, the plots provide for a qualitative overview.

For the clamped version, Figure 198, a comparison is made between hoop and axial stresses for the mechanical and hydraulic resonances encountered during the test. The hoop radial stresses average 25,000 psi while the longitudinal or axial stresses are about 11,000 psi. The reason the axial stress is not exactly half of the hoop stress is due to the additional influences of the axial load and the bending moment.

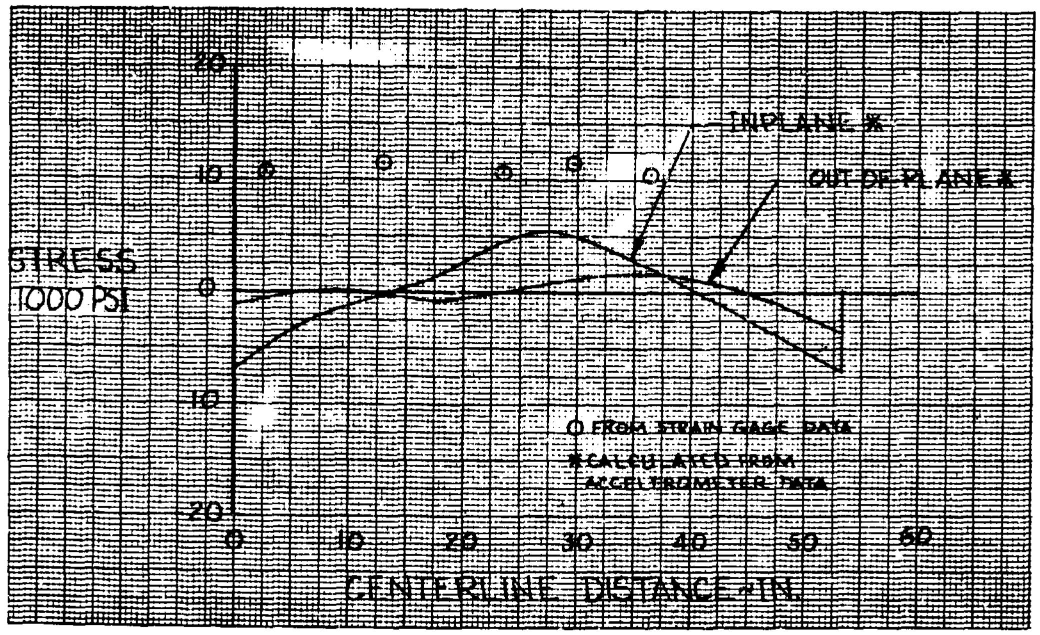


FIGURE 193 HYDRAULIC LINE MECHANICAL RESPONSE
ONE-ELBOW PIPE UNCLAMPED
STRESS COMPARISON
FREQUENCY 1030 HZ PUMP SPEED 1700 RPM

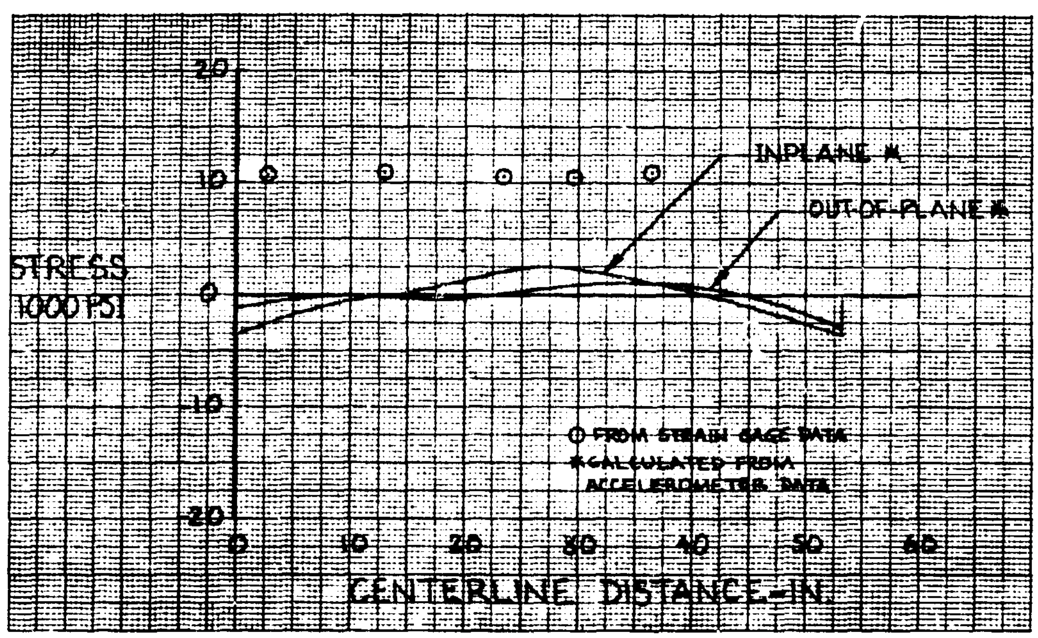


FIGURE 194 HYDRAULIC LINE MECHANICAL RESPONSE
ONE-ELBOW PIPE UNCLAMPED
STRESS COMPARISON
FREQUENCY 1030 Hz PUMP SPEED 2300 RPM

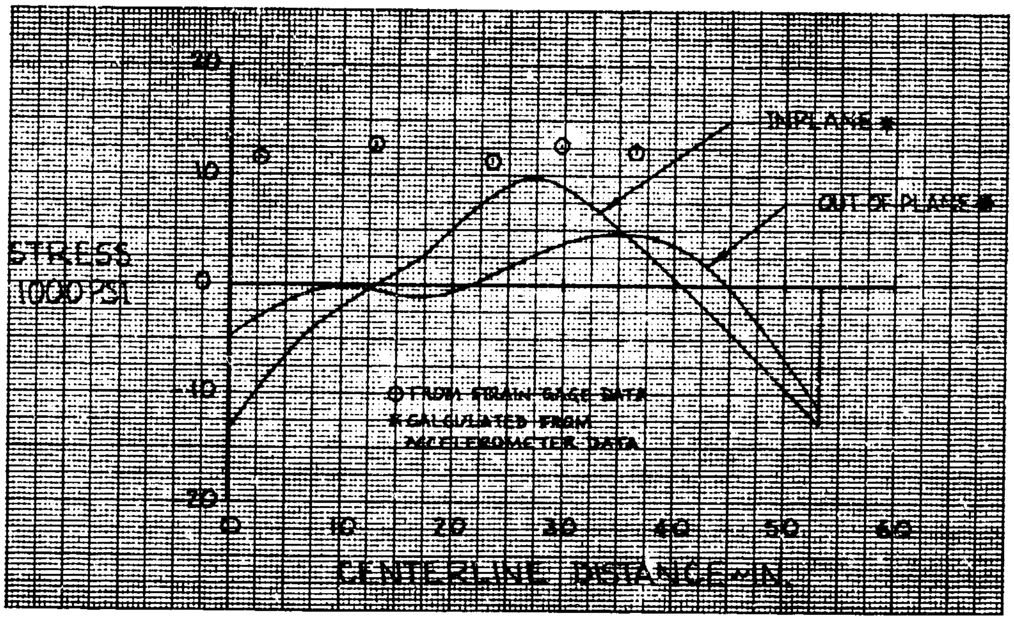


FIGURE 195 HYDRAULIC LINE MECHANICAL RESPONSE,
ONE-ELBOW PIPE UNCLAMPED
STRESS COMPARISON,
FREQUENCY 1030 RPM PUMP SPEED 3440 RPM

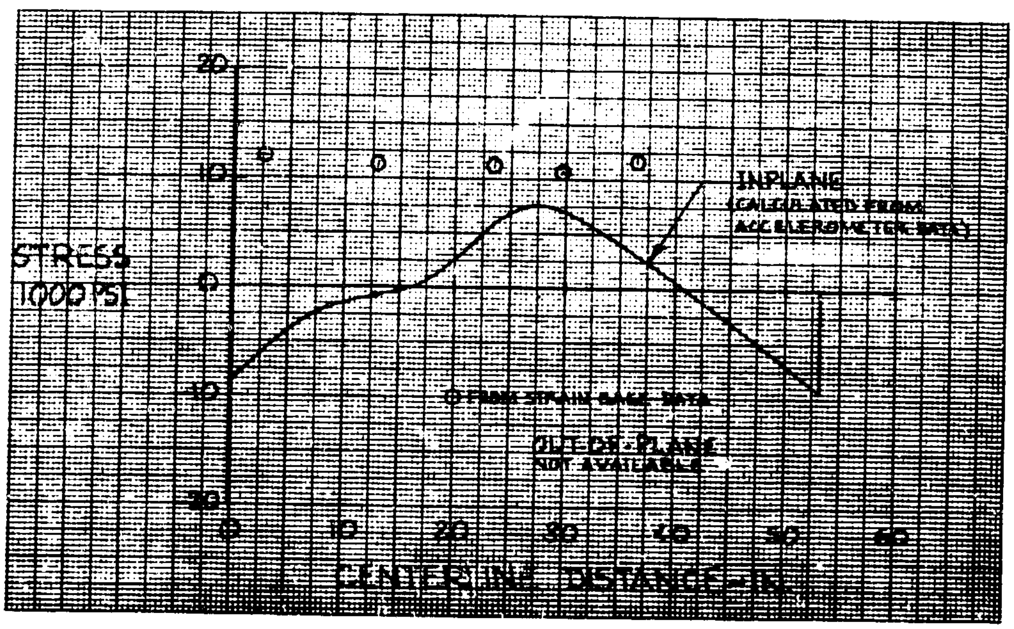


FIGURE 196 HYDRAULIC LINE MECHANICAL RESPONSE
ONE-ELBOW PIPE UNCLAMPED
STRESS COMPARISON
FREQUENCY 870 HZ PUMP SPEED 2900 RPM

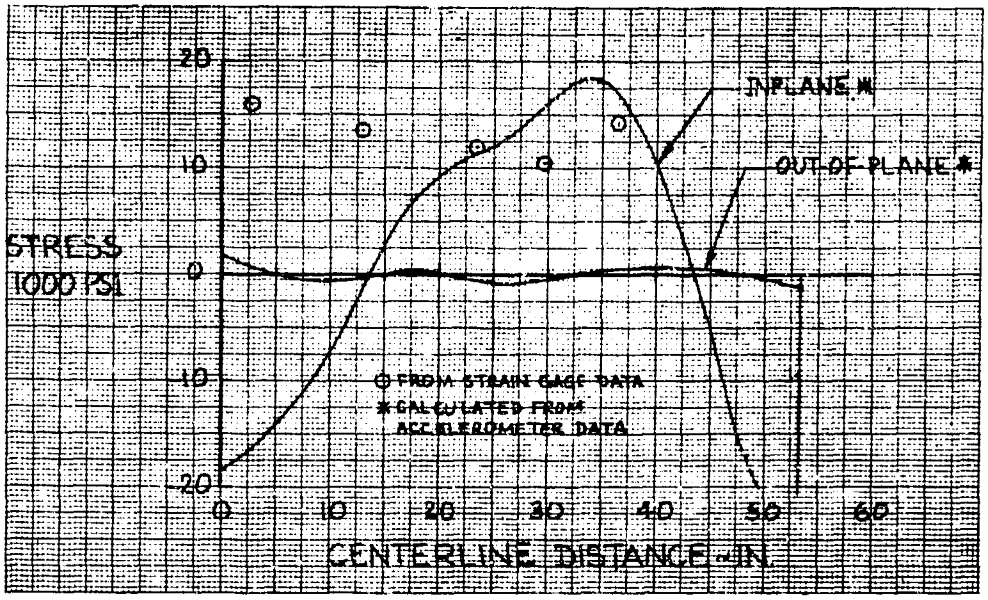


FIGURE 197 HYDRAULIC LINE MECHANICAL RESPONSE
ONE-ELBOW PIPE UNCLAMPED
STRESS COMPARISON
FREQUENCY 650 HZ PUMP SPEED 4300 RPM

RESONANCES	MECHANICAL			HYDRAULIC		
FREQUENCY (HZ)	400	490	570	240	870	650
PUMP SPEED (RPM)	2600	3250	3830	1600	2900	4300
HOOP STRESS (PSI)	24800	25000	25200	24600	25000	25000
AXIAL STRESS (PSI)	10900	11000	11600	10600	11600	11900

FIGURE 198 ONE-ELBOW PIPE CLAMPED
STRESSES BASED ON STRAIN MEASUREMENTS

e. Fatigue Life

The Goodman diagram developed for the one-inch outside diameter line with an .051-in. wall titanium is shown in Figure 199. The data is based on tests performed at MCAIR at 3000 psi. It must be noted that it is restricted to stresses adjacent to internally swaged Dynatube fittings. Stress levels, including installation stress, must stay below the 10^7 cycle line throughout the pump speed. The test results presented in Figures 197 through 198 show that the test specimen would have adequate fatigue life because the values fall below the referenced cycle line. This applies to both sets of data (accelerometer and strain gages) although the accelerometer data at the peak hydraulic resonance of 4300 rpm indicates marginal fatigue if the pump speed dwells at this resonance.

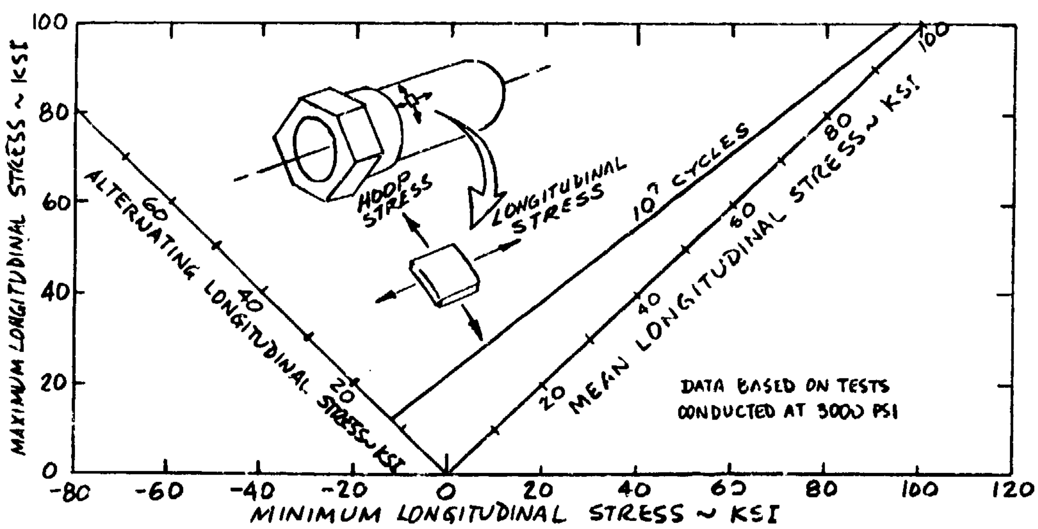


FIGURE 199 GOODMAN DIAGRAM: 1.00 x 0.051 TITANIUM
3A1-2.5V TUBING
APPLICABLE TO STRESSES ADJACENT TO INTERNALLY
SWAGED DYNATUBE FITTING

SECTION VII
SUMMARY AND CONCLUSIONS

1. HYTRAN PROGRAM

a. Empirical Pump Model

First order and second order HYTRAN empirical pump models were developed from test data. The mathematical descriptions are in a simple form which permits quick usage without requiring large amounts of input data. The models have the flexibility to allow the user to obtain any required degree of accuracy.

The computer program input data is dependent on measured test data. However, once the input parameters have been established for various volume systems and test conditions, they can be applied to other system simulations.

Both HYTRAN empirical pump models simulate outlet pressure and flow response to turn-on and turn-off transients. Good correlation has been obtained between computed output and measured test data.

b. Basic Pump Model

Changes were made to the basic HYTRAN pump model to improve correlation with test data and enhance model usability. Program changes involved an orifice coefficient calculation for the compensator valve, a steady state flow force calculation, rederivation of actuator pressure allowing for diametral clearance leakage past the compensator, case drain flow area calculation, and hanger offset flow calculation.

Input data requiring computation was derived and examples were given defining piston acceleration forces and spring forces on the hanger and a hanger damping term.

Turn-on and turn-off transient simulations were made using the improved basic pump. Results of the computer simulations correlated well with test data.

c. Lossless Line Model

In the present version of the HYTRAN program, a fix-up is made in the velocity of sound whenever a line connecting two components has a length less than a Δx. The modification makes the line a Δx length and uses only the static friction effects in the transient simulation. A timing error exists between the computed pressure/flow at the inlet and outlet of the line. But for small Δt's the effect is oftentimes negligible.

Several approaches were tried to connect closely coupled components without creating a timing error. One approach was to solve the components individually by using artificial characteristic pressure and impedance terms. This technique did not work.

An analytical approach was tried to rewrite the HYTRAN component models to allow them to be directly connected and then simultaneously solve for pressures, flows, and component variables. This procedure was costly and time consuming to implement. Other methods, involving iteration procedures to solve the close coupled component and line system using upstream and downstream characteristic pressures, were also prohibitive in cost. A more attractive solution was to use the port energy method. Implementing this concept was not within the scope of the present contract, as it would have required extensive rework of the HYTRAN program.

The most practical approach to decreasing the timing errors was to create a lossless line model. The lossless line model removes the line friction and calculates the fluid velocity based on fluid properties, and gives reasonable results.

For the present HYTRAN program, the simplest approach to minimizing errors caused by closely coupled components is to reduce the calculation time interval. Where computer machine cost is a factor,and the connection distance is short, use the lossless line model.

d. HYTRAN Program Improvements

The two way control valve, check valve, priority valve and restrictor models were improved. A reservoir level sensing bootstrap reservoir model and a dual system, tandem, valve controlled actuator model were added to the program. Several enhancements were made to the STORE, GRAPH, BLOCK DATA and COMP sub-programs, with the appropriate user manual changes. In addition, a second order response function, SECORD was added to the utility library.

2. HSFR PROGRAM

a. Amplitude Prediction

Substantial improvement has been made in the computation of peak pressure pulsations. The key to better correlation with test data was found in the accurate computation of pump flows and rederivation of the dynamic pressure-flow balance in the pump model.

Pump outlet temperature and print plot locations were varied to show the sensitivity of these parameters in finding the peak pressure location in the test system at any given RPM.

b. Attenuation Techniques

Two general approaches to solving pressure pulsation problems were presented. The first technique relied on devices that shifted the unwanted pulsations

outside the using range of the system. The second technique involved components that actually reduced the pressure pulsation energy at the critical resonant frequencies.

c. Empirical Pump Model

The repeated failure of the hydraulic acoustic generator prohibited further development and verification of the empirical pump model. However, pump flow and shunt impedance can be computed from values calculated by the complete pump model and used to define an empirical model for subsequent HSFR simulations.

d. Output Options

HSFR printed output capabilities have been greatly extended. The user has the option of printing critical computational variables and plotting a standing wave for any line in the simulation.

3. SSFAN PROGRAM

a. Quasi-Transient

The addition of the Quasi-Transient calculation extends the capability of SSFAN severalfold. The size of the time step selected for calculation could affect the accuracy of the calculation if it is too large. For the limited usage that it has to date, it has supplied reasonable results for predicting the operating times of subsystems.

b. Program Improvements

The many program changes provide a simpler internal program for assembling systems and the new matrix calculation technique provides a faster (lower cost) solution. This has been achieved while keeping the input data format the same for most component elements which should result in minimal problems for users who switch to the new version.

A flow regulator (restrictor) was developed to keep the flow constant in a leg. An orifice sizer model was also developed from this model to compute the orifice size at the input flow rate.

4. HYTTHA Program

a. Program Improvements

Several improvements were made to the HYTTHA program. The user has the ability to default to various structural, wall, or fluid temperatures throughout the system. The system can also be divided into different environments to simulate actual conditions and through the constant pressure reservoir subroutine, the user can specify a fluid temperature time history.

The pump model was enhanced and a start-up capability was added. The computational subroutines were modified to include a compressed matrix method that reduced the storage space required to solve for the steady state pressure values.

A bend energy loss pressure drop was added to the line subroutine, replacing the equivalent length method.

b. Verification Tests

Two thermal transient test runs were simulated using the HYTTHA program. The predicted temperature values were generally 20 degrees higher than the measured data at the end of the computer run. Much of the error is attributable to the thermal pump model. Without the pump model in the simulation, and using the measured fluid temperature at the pump as a boundary condition, the HYTTHA computer simulation gave better correlation to the test data.

5. HLMR Program

The second series of test on the one-elbow specimen using strain gages reconfirmed the previous mechanical and hydraulic resonances measurements made with accelerometers. The use of a stiffer clamp resulted in lower mechanical frequencies such that the resonances were within the operating regime of the pump.

The stresses based on strain measurements indicate, by means of a Goodman diagram, that specimens of this size have adequate fatigue life.

Investigation into the use of the NASTRAN finite element method program and other existing MCAIR programs including the HSFR program indicates that it is feasible to derive a more accurate response calculation scheme.

SECTION VIII
RECOMMENDATIONS

1. HYTRAN Program

a. Empirical Pump Model

A provision can be made to model reversal transients using the empirical pump model. The feasibility of adding case drain and inlet models should be investigated.

b. Basic Pump Model

Further modeling work needs to be accomplished to improve the inlet and case drain sections of the pump model. Test data taken using the F-4 Abex pump will be used to verify the F-4 basic pump model. Pump damping characteristics will be further investigated from the empirical pump test data to arrive at appropriate viscous damping expressions for use in the pump model.

c. Program Improvements

Program improvements include further verification and simplifcation of existing models, new component models and enhanced programming features.

(1) Actuator Model - A table look-up is needed for the load/stroke curve. Further verification testing can include the effects of various loads on stiction and dynamic friction.

(2) Accumulator Model - The current model assumes a perfect gas law relationship for accumulator operation. A user selectable specific heat ratio to suit the type of application being analyzed needs to be incorporated.

(3) Three Way Valve Model - A two position three way valve model needs to be written to complement the existing two-way two position and four-way three position valve models.

(4) Line Cavitation Model - An improved HYTRAN line cavitation mode is required. This eliminates the need to write cavitation models for many of the components.

(5) Multiple Plot Option - Modified output plotting routines are required to make plots of different parameters on a single graph. This would provide an easy way to compare various data parameters.

(6) Steady State Solution Technique - The sparse matrix equation solver used in HYTTHA and SSFAN can be implemented in the HYTRAN program. The payoff is significant cost reduction for program usage.

2. HSFR PROGRAM

Having a working hydraulic acoustic generator (HAG) allows fundamental and extremely useful research into frequency related phenomena in hydraulic systems. This HAG would:

- o Provide a source of measurable dynamic flow
- o Allow accurate component model verification with a harmonic free acoustic signal of variable frequency
- o Allow endurance testing to a specific acoustic environment
- o Allow accurate measurement of load circuit impedance which can be used to determine pump shunt impedance
- o Provide calibration reference for evaluation of dynamic flowmeters

The HAG unit requires work. A complete bearing redesign is recommended. In addition to the above items the HAG can be used to determine line damping factors at typical hydraulic system operating frequencies.

3. SSFAN PROGRAM

a. Thermal Prediction

For design purposes, a reasonable estimation of fluid temperatures has proven to be desirable. It would appear that a simple method for calculating the thermal rise in a hydraulic system could be developed and used with the quasi-transient program. Practical experience has shown that calculated conduction, convection and radiation heat losses from the fluid are less than actually seen in aircraft hydraulic systems. It is recommended that an initial study be made to look at actual aircraft and iron bird thermal test data and determine the feasibility for developing a general heat dissipation coefficient that would give reasonable temperature rise results.

b. Waterhammer Prediction

An addition can be made to the quasi-transient program to estimate the waterhammer pressure rise due to valve closings and bottoming of actuators. The actual leg lengths are already stored in the program. A test would be made to determine whether the valve was fast closing or slow closing using the "2 x length to pressure source ÷ velocity of sound" criteria and calculate the pressure rise using the appropriate equation.

4. HYTTHA Program

Further verification needs to be accomplished on the pump model. Additional verification work involving actual aircraft systems is also desirable.

5. HLMR PROGRAM

For near term activities it is recommended that development be made of wide band pulsation attenuators. The lowering of pump pulsation will correspondingly lower the line responses nearest the pump. To determine the effect on lines testing should be made with attenuators on the previously tested specimens for direct comparison, using accelerometers.

For long term activities it is recommended that a general purpose computer program be developed using present MCAIR analytical capabilities. The objective would be to couple the HSFR output (forging function) with a Nastran Program (which accurately predicts mode shapes and deflections) and an existing stress prediction program. The stresses due to pulsations AND allowable misalignment would be determined. Since infinite fatique life is required, the program would be used to identify unacceptable stress levels and the optimum solution in achieving infinite life.

APPENDIX A

PUMP MODEL

HSFR TECHNICAL MANUAL (AFAPL-TR-76-43 VOL. IV)

4.0 PUMP SUBROUTINE

4.1 INTRODUCTION AND FLOW DIAGRAM

Subroutine PUMP is a general, detailed model of a rotating, axial, nine-piston, pressure-compensated hydraulic pump. The model computes the ability of the pump to deliver flow against an output pressure by modeling the non-linear relationship between pump output flow and pressure in the time domain. The main program calculates the harmonic load impedance of the rest of the circuit, and this provides the linear phase and gain relationship between the harmonic flows into the load and the corresponding pressures across the load, in the frequency domain. The balance is obtained in the time domain, although a check is performed in the frequency domain. The model can also calculate inlet flow based on the return system load impedance which is calculated by the main program.

PUMP considers valving areas, pre-compression, de-compression, steady state swash angle or pressure, fluid bulk modulus, pump internal leakage, circuit termination flow, and piston motion. Steady state swash angle is calculated and balanced as a function of pump internal leakage, circuit over-board leakage, and pump speed. If the swash angle is maximum, the steady state pressure is calculated. Swashplate flexibility and swashplate-compensator dynamics are not included.

Piston pressure at the beginning of pre-compression is assumed constant and equal to the inputed steady state inlet pressure if inlet side analysis is not selected. Otherwise, piston pressure is calculated continuously for the full pump revolution.

Figure 4-1 is a general flow chart of the PUMP subroutine. The specification section includes initialization of variables from input data, and calculates several constants. Specification statements are followed by

initialization of pump variables from the main program input data, and calculation of valve port areas for a full 360° revolution. These operations are performed only once, when PUMP is called on the first pump speed. Steady state swash angle or outlet pressure are then calculated, followed by the precompression pressures. Pump outlet flow is calculated and then analyzed by Fourier analysis. The steady state component of the Fourier analysis is then compared to the total steady state circuit flow. If these differ excessively, the swash angle or outlet pressure are corrected and the pump outlet flow is recalculated. When the swash angle or outlet pressure produce a steady state flow essentially equal to the pump and circuit termination leakage flows, the Fourier analysis is completed to calculate harmonic flows up through the harmonic of interest. Harmonic pressure and flow are then balanced dynamically by reconstructing the time dependent output pressure and recomputing flow from section 4. If inlet analysis is not selected, pump outlet flow and pressure for the harmonic of interest are returned to the main program.

If inlet analysis is selected, piston decompression and inlet flow is calculated. If return system analysis is selected, Fourier analysis of inlet flow is performed, followed by dynamic balancing of inlet flow with the return system load. Otherwise, inlet flow is based on the constant steady state inlet pressure and the program goes directly to the hanger torque calculations. Inlet and outlet flow and pressure for the harmonic of interest are then returned to the main program.

The PUMP subroutine is divided into thirteen sections. Each section is discussed and a listing of that section is presented individually in subsequent paragraphs.

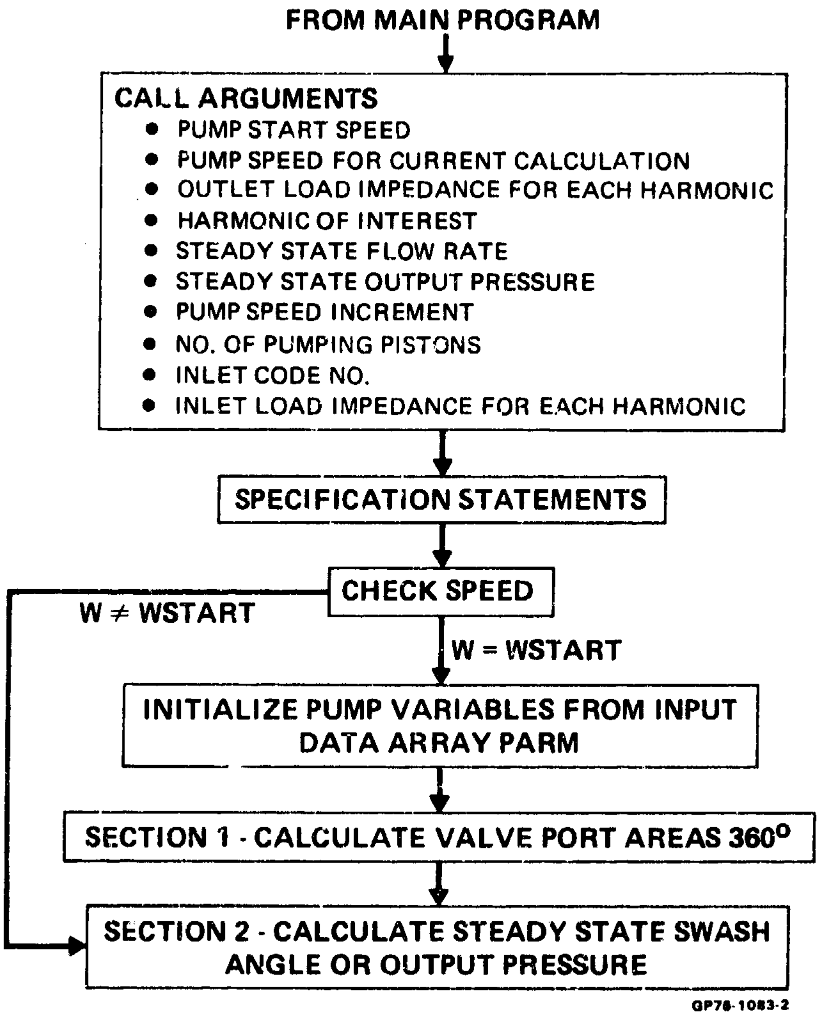


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FIGURE 4-1
HSFR COMPUTER PROGRAM - PUMP SUBROUTINE FLOW CHART

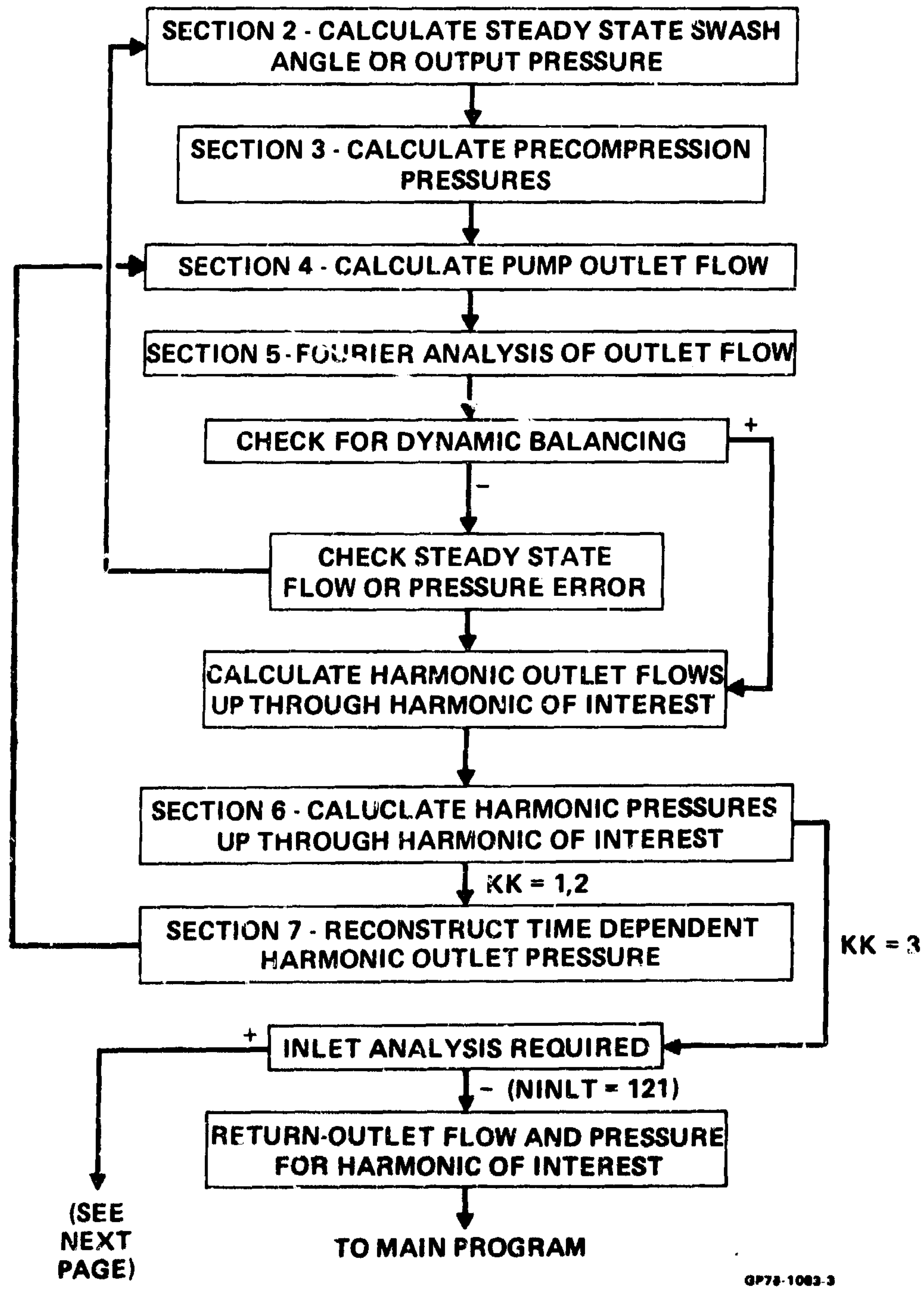

FIGURE 4-1 (Continued)
HSFR COMPUTER PROGRAM - PUMP
SUBROUTINE FLOW CHART

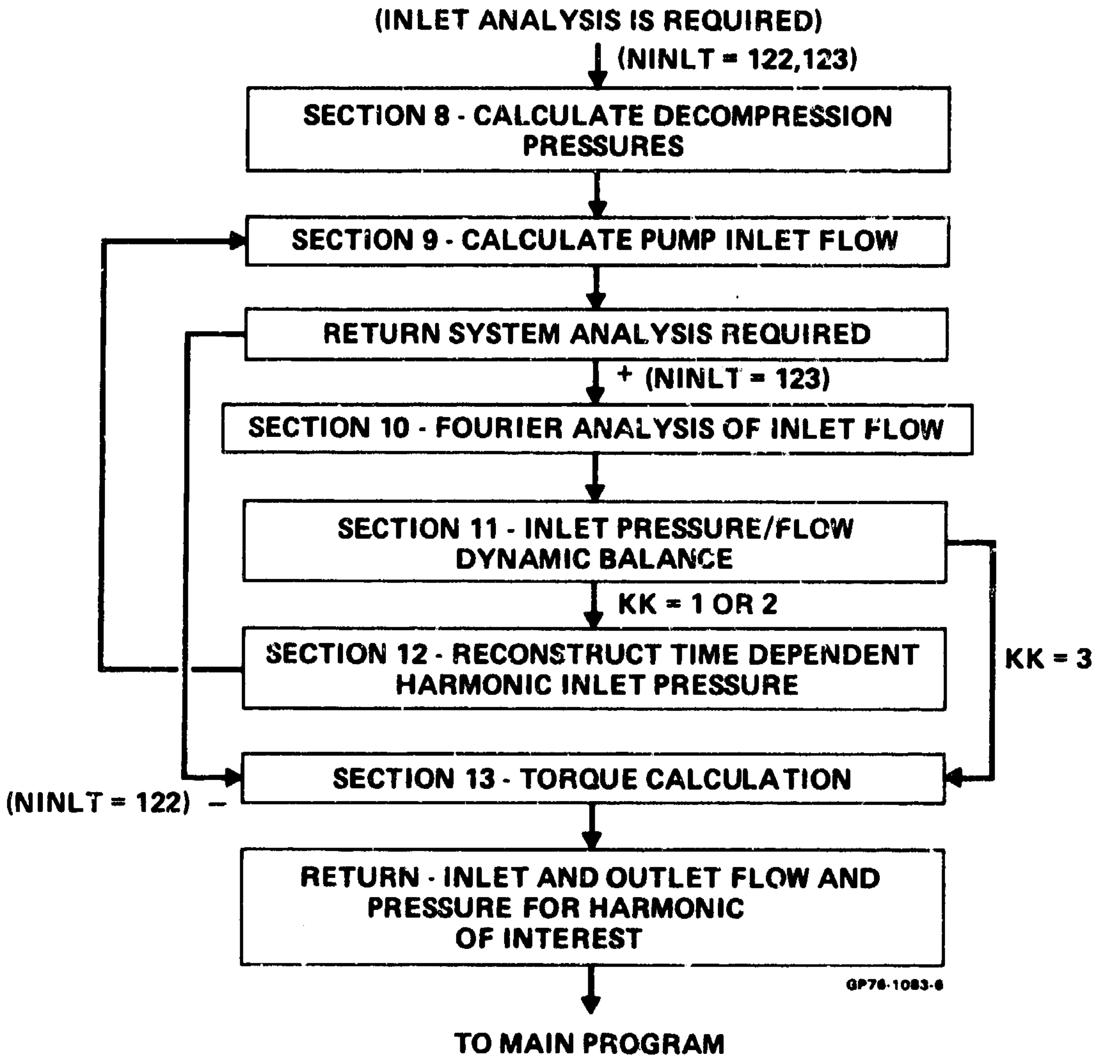

FIGURE 4-1 (Continued)
HSFR COMPUTER PROGRAM - PUMP
SUBROUTINE FLOW CHART

4.1.1 Math Model - See sub-paragraphs in section 4.

4.1.2 Assumptions - See sub-paragraphs in section 4.

4.1.3 Computation Method - See sub-paragraphs of section 4.

4.1.4 Approximations - See sub-paragraphs of section 4.

4.1.5 Limitations - See sub-paragraphs of section 4.

4.1.6 Variable Names - Variable names unique to the PUMP subroutine are listed below. Common variables are discussed in the main program paragraph 3.1.6.1.

SYMBOL	DESCRIPTION	UNITS
ACTLEV	Swashplate actuator lever arm	IN
ACTLEVO	Swashplate actuator lever arm at zero angle	IN
AHLEN	Temporary variable used in valve area calculation	-
AINC	Incremental shaft rotation angle	DEG
AINDEX,ALPHA1, ALPHA2,ALPHA3, ALPHA4,ALPHA5, ALPHA6,ALPH2A	Temporary variables used in valve area calculation	-
AN,N	Angular spacing between pistons used in Fourier Analysis	DEG
ANGCR	Input data - swashplate fixed cross angle	DEG
ANGLE,ANGLE1, ANGLE2	Temporary variables used in valve area calculation	-
ANPRCL,ANPRSC, ANPRSO,ANSUCL, ANSUOP,ANSUSC, ANSUSO	Angular position of valve area index	RAD
ARACT	Area of swashplate actuator	IN**2
AREA	Valve opening area	IN**2
AREA1,AREA2, AREA3,ASUB1, ASUB1A,ASUB2	Temporary variables used in valve area calculation	-
ASWASH	Swashplate variable angle	RAD
ASWERR(IFL)	Swashplate angle computed during IFL iteration	RAD
AT	Angular position of pistons	RAD

4.1.6 (Continued)

SYMBOL	DESCRIPTION	UNITS
ATOFF	Angle of swashplate offset	DEG
ATPR,ATSU	Temporary variables used in torque calculation	-
BLEN	Temporary variable used in valve area calculation	-
BULKP	Bulk modulus of fluid at working pressure	PSI
C,COEF,C1	Temporary variables used in Fourier Analysis	-
CAVOL,CAVOLD	Piston cavitation volume	IN**3
CD	Orifice coefficient	
CPRESS	Input data - case pressure-steady state	PSI
CSPRESS	Input data - case to inlet pressure at zero case drain flow	PSI
DELPX	Differential pressure between piston chamber and pump inlet/outlet	PSI
DEQUIV	Equivalent diameter of cylinder slot flow area	IN
DIACT	Input data - swashplate actuator diameter	IN
DIAM	Equivalent diameter of pressure slot area	IN
DIAPIS	Input data - pumping piston diameter	IN
DISACT	Actuator displacement	IN
DISAM	Maximum actuator displacement	IN
DLEAK	Leakage from one cylinder during rotation through incremental angle AINC	IN**3
DPLEAK	Differential pressure in cylinder due to leakage during precompression/decompression	PSI
DRATIO	DEQUIV/DIAM	-
DSWERR	Incremental swashplate angle	RAD
DT	Incremental time for rotation through incremental angle AINC	SEC

4.1.6 (Continued)

SYMBOL	DESCRIPTION	UNITS
DVOL	Incremental volume change of piston for rotation through incremental angle AINC	IN**3
DX	Incremental piston stroke for rotation through incremental angle AINC	IN
ETA	Not used	-
FLERR(IFL)	Steady state flow error computed during IFL iteration	CIS
FNTZ	Temporary variable used in Fourier Analysis	-
FQ1(-,-)	Complex dynamic output flow	CIS
FQ1I(-,-)	Complex dynamic input flow	CIS
HLEN	Temporary variable used in valve area calculation	-
HOFF	Input data - swashplate offset	IN
HPRERR(IFL)	Steady state output pressure computed during IFL iteration	PSI
HPRESS	Steady state output pressure	PSI
I,J,M,N	Integer counter	-
IFL	Integer counter for number of steady state balance loop iterations	-
INDEX1,INDEX2, INDEX3,INDEX4	Temporary index values used in valve area calculation	-
IOPT( )	Input data - write option array	-
KINTRP	Integer counter for number of interpolation loops used in steady state balance calculation	-
KK	Integer counter for dynamic balancing test	-
KKN	Order of harmonic	-
LEAK	Constant for piston lap leakage	CIS/PSI
LK1-LK6	Temporary variables used in flow calculation	-
LPRESS	Input data - steady state inlet pressure	PSI

4.1.6 (Continued)

SYMBOL	DESCRIPTION	UNITS
NAPP	Number of active pistons pumping	-
NAPS	Number of active pistons sucking	-
NBEG, NBEG1-NBEG8	Integer DO LOOP counter	-
NDEG	Integer counter for stepping cylinder block rotation beginning with NDEG = 1 at piston bottom dead center on the swashplate	-
ND9	Maximum value of NDEG	-
NHARM	Integer form of WHARM	-
NINLT	Inlet identification code number	-
NKM	Index position of active pumping or sucking piston	-
NOPIST	Integer form of PISTNO	-
NPRCL	Last index position (NDEG) before cylinder slot is fully closed to pressure slot	-
NPROP	Last index position (NDEG) before cylinder slot is fully open to pressure slot	-
NPRSCL	Last index position (NDEG) before cylinder slot starts to close to pressure slot	-
NPRSOP	Last index position (NDEG) before cylinder slot starts to open to pressure slot	-
NSTART	Integer counter for start of calculation	-
NSUCL	Last index position (NDEG) before cylinder slot is fully closed to suction slot	-
NSUOP	Last index position (NDEG) before cylinder slot is fully open to suction slot	-
NSUSCL	Last index position (NDEG) before cylinder slot starts to close to suction slot	-
NSUSOP	Last index position (NDEG) before cylinder slot starts to open to suction slot	-
ORF	Not used	
PACTU	Actuator pressure used in torque calculation	PSI

4.1.6 (Continued)

SYMBOL	DESCRIPTION	UNITS
PHI1,PHI2,PHI4	Temporary variables used in valve area calculation	-
PIA	Piston area	IN**2
PIMASS	Piston mass	LB-SEC**2/IN
PISPR( )	Piston pressure	PSI
PISTNO	Input data - number of pistons	-
PLEAK	Leakage out of cylinder	CIS/PSI
POVOL	Input data - cylinder volume at midstroke	IN**3
PPI( )	Time dependent amplitude of pump inlet pressure	PSI
PPM,PPP	Not used	PSI
PPT( )	Time dependent amplitude of pump outlet pressure	PSI
PQ1 (-,-)	Complex output test pressure in dynamic balancing	PSI
PQ1I(-,-)	Complex inlet test pressure in dynamic balancing	PSI
PRESS	Input data - steady state pump output pressure	PSI
PSPG	Actuator pressure due to spring force	PSI
P3	Complex test pressure used in constructing time dependent inlet/outlet dynamic pressure	PSI
QIFC( )	COSINE peak amplitude of pump dynamic inlet flow	CIS
QIFS( )	SINE peak amplitude of pump dynamic inlet flow	CIS
QIN	Flow into piston	CIS
QMAXI	Maximum steady state pump outlet flow at maximum swash angle and zero outlet pressure at maximum RPM	CIS
QMAXIY	Maximum steady state pump outlet flow at maximum swash angle and zero outlet pressure at given RPM	CIS

4.1.6 (Continued)

SYMBOL	DESCRIPTION	UNITS
QMAXY	Maximum steady state pump putlet flow at maximum swash angle and input steady state pressure at given RPM	CIS
QMECH	Flow out of cylinder due to piston displacement	CIS
QMMAX	Maximum flow out of cylinder due to piston displacement at given swash angle	CIS
QOUT	Flow out of piston	CIS
QOVB	Total overboard flow from main program	CIS
QOVBT	Not used	-
QQFC( )	COSINE peak amplitude of pump dynamic outlet flow	CIS
QQFS( )	SINE peak amplitude of pump dynamic outlet flow	CIS
QQI( )	Time dependent amplitude of inlet flow	CIS
QQT( )	Time dependent amplitude of outlet flow	CIS
QWBEG	Maximum steady state pump outlet flow at maximum swash angle and input steady state pressure at starting RPM	CIS
QWEND	Maximum steady state pump outlet flow at maximum swash angle and input steady state pressure at maximum RPM	CIS
QZERO	Straight line extrapolation of QWEND and QWBEG to zero RPM	CIS
RBORC	Input data - cylinder centerline circle radius	IN
REN	Calculated Reynolds number	-
RV	Input data - cylinder and valve plate slot centerline circle radius	IN
R1	Input data - cylinder slot end radius	IN
R2	Input data - valve plate pressure slot end radius	IN
R4	Input data - valve plate suction slot end radius	IN

4.1.6 (Continued)

SYMBOL	DESCRIPTION	UNITS
R1SQ,R2SQ	Values used to calculate equivalent pressure slot area diameter	-
S	Temporary variable used in Fourier Analysis	-
SA	Half amplitude of piston stroke due to swash angle	IN
SAM	Maximum piston half stroke at maximum swash angle	IN
SLEAK	Leakage into cylinder bore	CIS/PSI
SLOTW	Input data - cylinder slot width	IN
SLTHAG	Cylinder slot width half angle	RAD
SSA	Half amplitude of piston stroke due to fixed swash plate cross angle	IN
SWASH	Input data - maximum swash angle	DEG
S1	Temporary variable used in Fourier Analysis	-
THECON	Constant used in time dependent pressure calculation	-
THETA	Angle of cylinder block rotation	RAD
THETA1,THETA2, THPRE1,THPRS1, THSUC1,THSUE1	Constants used in valve area calculation	-
THPRE	Input data-valve plate pressure slot end angle	DEG
THPRS	Input data-valve plate pressure slot start angle	DEG
THSUCE	Input data-valve plate suction slot end angle	DEG
THSUCS	Input data-valve plate suction slot start angle	DEG
TLEAK	Input data-pump internal leakage	CIS
TLEAKT	Not used	-

4.1.6 (Continued)

SYMBOL	DESCRIPTION	UNITS
TORAAV	Average total torque due to piston inertia	IN-LB
TORPAV	Average total torque due to piston pressure	IN-LB
TORQ	Total swash plate torque at each calculation point	IN-LB
TORQAC	Swashplate torque due to piston inertia at each calculation point	IN-LB
TORQAS	Summation of TORQAC	IN-LB
TORQPR	Swash plate torque due to piston pressure at each calculation point	IN-LB
TORQPS	Summation of TORQPR	IN-LB
TORQSU	Summation of TORQ	IN-LB
TORQ1	Temporary variable used in torque calculation	-
TORTAV	Total average torque	IN-LB
TRACAV	Average torque due to cross angle	IN-LB
TRQACR	Torque due to cross angle at each calculation point	IN-LB
TRQACS	Summation of TRQACR	IN-LB
U0,U1,U2,U3	Temporary variables used in Fourier Analysis	-
VA,VALEAK	Cylinder volume at given piston position	IN**3
VAREA(NDEG)	Valve-area at NDEG index position	IN**2
W	Harmonic frequency	RAD/SEC
XANG	Index angle for calculation of QMECH	RAD
XLAST,XNEW	Piston position	IN
XLEN	Temporary variable used in valve area calculation	-
Y	Pump speed	RPM
YOPT( )	Speed for write option output	RPM
ZAP( )	Complex impedance of load on pump inlet	PSI/CIS
ZE	Temporary variable used in dynamic flow balance calculation	-

4.1.6 (Continued)

SYMBOL	DESCRIPTION	UNITS
ZIP( )	Complex impedance of load on pump outlet	PSI/CIS
ZO	Pump shunt impedance	PSI/CIS

4.1.7 Specifications and Initialization-Listing

```
      SUBROUTINE PUMP (WSTART,Y,ZIP,NHARM,QOV3,PRESS,WINC,PISTNO,
     +NINLT,ZAP WEND,IOPT,YOPT)
C
C * REVISED MARCH 21, 1977 *
C
C         *VARIABLE TYPES, DIMENSIONS, COMMONALITY*
C
      REAL LPRESS,LEAK,LK1,LK2,LK3,LK4,LK5,LK6
      COMPLEX BETA,G,P,Q,Z
       COMPLEX ZO,ZIP,P3,ZAP,ZE
      COMPLEX FQ1,PQ1,PQ1I,FQ1I
      COMMON BETA,G,P,Q,Z, XBE,BER,BEI,BERP,BEIP,RHO,BULK,VOL,W,VISC,PAR
     1M PI, IEL,NEL,KTYPE(40)
      DIMENSION G(2,2,40),PARM(8,40),P(40),Q(40),Z(40)
      DIMENSION QQT(81),PPT(81),QQFC(11),QQFS(11)
      DIMENSION ZIP(10),ZAP(10)
      DIMENSION VAREA(800)
      DIMENSION FQ1(10,3),PQ1(10,3)
      DIMENSION PISPR(800),QQI(81),PPI(81)
      DIMENSION QIFC(11),QIFS(11),FQ1I(10,3),PQ1I(10,3)
      DIMENSION FLERR(10),HPRERR(10),ASWERR(10)
      DIMENSION IOPT(7),YOPT(4)
      DATA VAREA/800*0.0/,PISPR/800*0.0/
C
C                       *SOLUTION METHOD*
C
C *INITIALIZE VARIABLES FROM INPUT DATA OR MAIN PROGRAM*
C
      IF(Y.NE.WSTART) GO TO 140
      R1=PARM(1,1)
      SLOTW=PARM(2,1)
      RV=PARM(3,1)
      RBORC=PARM(4,1)
      DIAPIS=PARM(5,1)
      POVOL=PARM(6,1)
      R2=PARM(7,1)
      R4=PARM(1,NEL+1)
      SWASH=PARM(2,NEL+1)
      TLEAK=PARM(3,NEL+1)
      ANGCR=PARM(4,NEL+1)
      THPRS=PARM(5,NEL+1)
      THPRE=PARM(6,NEL+1)
      THSUCS=PARM(7,NEL+1)
      THSUCE=PARM(8,NEL+1)
      LPRESS=PARM(1,NEL+2)
      HOFF=PARM(2,NEL+2)
      DISAM=PARM(3,NEL+2)
      ACTLEVO=PARM(4,NEL+2)
      PIMASS=PARM(5,NEL+2)
      CPRESS=PARM(6,NEL+2)
```

4.1.7 (Continued)

```
      CSPRES=PARM(7,NEL+2)
      DIACT=PARM(3,NEL+2)
      ARACT=DIACT**2*PI/4.
      ASWASH=SWASH/57.3
      NOPIST=PISTNO
```

4.2 SECTION 1 - VALVE AREA CALCULATION

Figures 4-2, 4-3, and 4-4 illustrate modeling parameters for a typical aircraft rotating piston hydraulic pump, including those required for area calculations.

Section 1 is a generalized computation of the valve flow area for one cylinder at every 1/2 degree position in a full revolution of the cylinder barrel. Area calculation is done once, the first time PUMP is called by the main program for the starting pump speed (W = WSTART). Computed valve areas for each 1/2 degree position are stored in the VAREA array. Valve areas and indexes are used in PUMP for outlet and inlet flow, and precompression and decompression calculations.

Section 1 first initializes the angular increment AINC to 1/2 degree of cylinder block rotation. Cylinder slot constant angles are calculated, as are the maximum number of pistons exposed to the pressure and suction slots for use in subsequent calculations.

Pressure slot valve areas are calculated beginning with the cylinder centerline at bottom dead center of the controlled swash plate angle (THETA = 0.0, NDEG = 1). Index angles are calculated to establish index positions which, in turn, control DO LOOP calculations of valve areas for each portion of cylinder revolution.

4.2.1 Math Model - The basic valve area calculation model is illustrated in Figure 4-5 for the opening of the cylinder slot to the pressure slot. The equations developed in Figures 4-5(a) through 4-5(g) assume that (R1.GE.R2). The relationships for the reverse situation are similar, except for calculation of the index angles. The differences are provided for in each DO LOOP where tests for (R2.GT.R1) are used to direct the flow of program execution to the appropriate calculation. The pressure slot closing calculations are executed in the same manner, progressing from fully open through each INDEX to fully closed. Suction slot valve areas

4.2 (Continued)

are also calculated with the math model shown in Figure 4-5, except that R4 replaces R2 and the main flow of program execution assumes that (R1.LT.R4).

4.2.2 Assumptions - None

4.2.3 Computation Method - Areas are calculated for each 1/2 degree increment of cylinder block rotation from 0 to 360 degrees.

4.2.4 Approximations - The small area ASUB1A shown on Figure 4-5(d) is approximated as the area of a triangle with base BLEN and height HLEN. The resulting error is considered to be insignificant.

4.2.5 Limitations - Pumps with non-typical valve plate configurations may require that valve areas be computed outside of PUMP, and be read into the main program.

4.2.6 Variable Names - See paragraph 4.1.6.

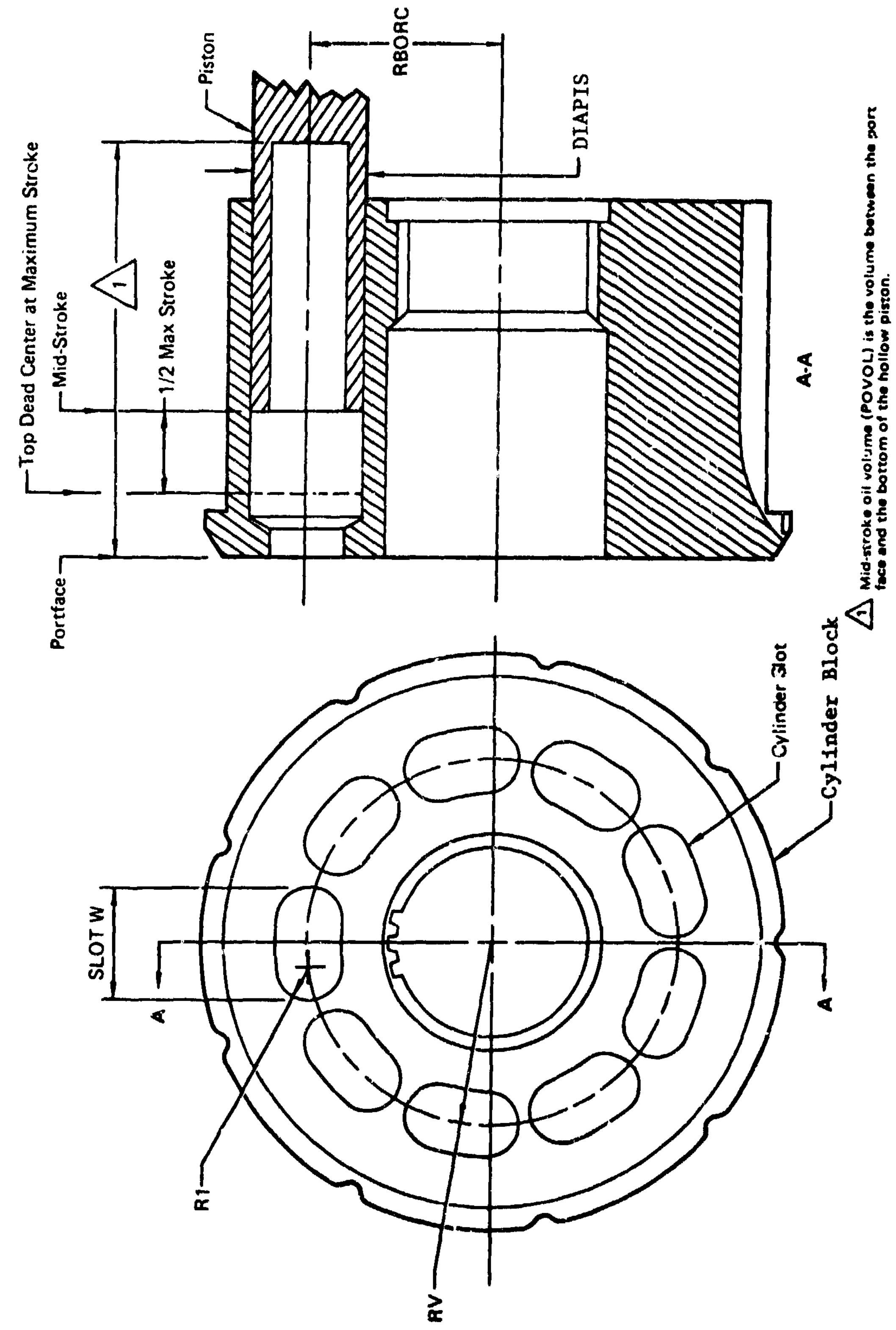

FIGURE 4-2
PUMP CYLINDER BLOCK PARAMETERS

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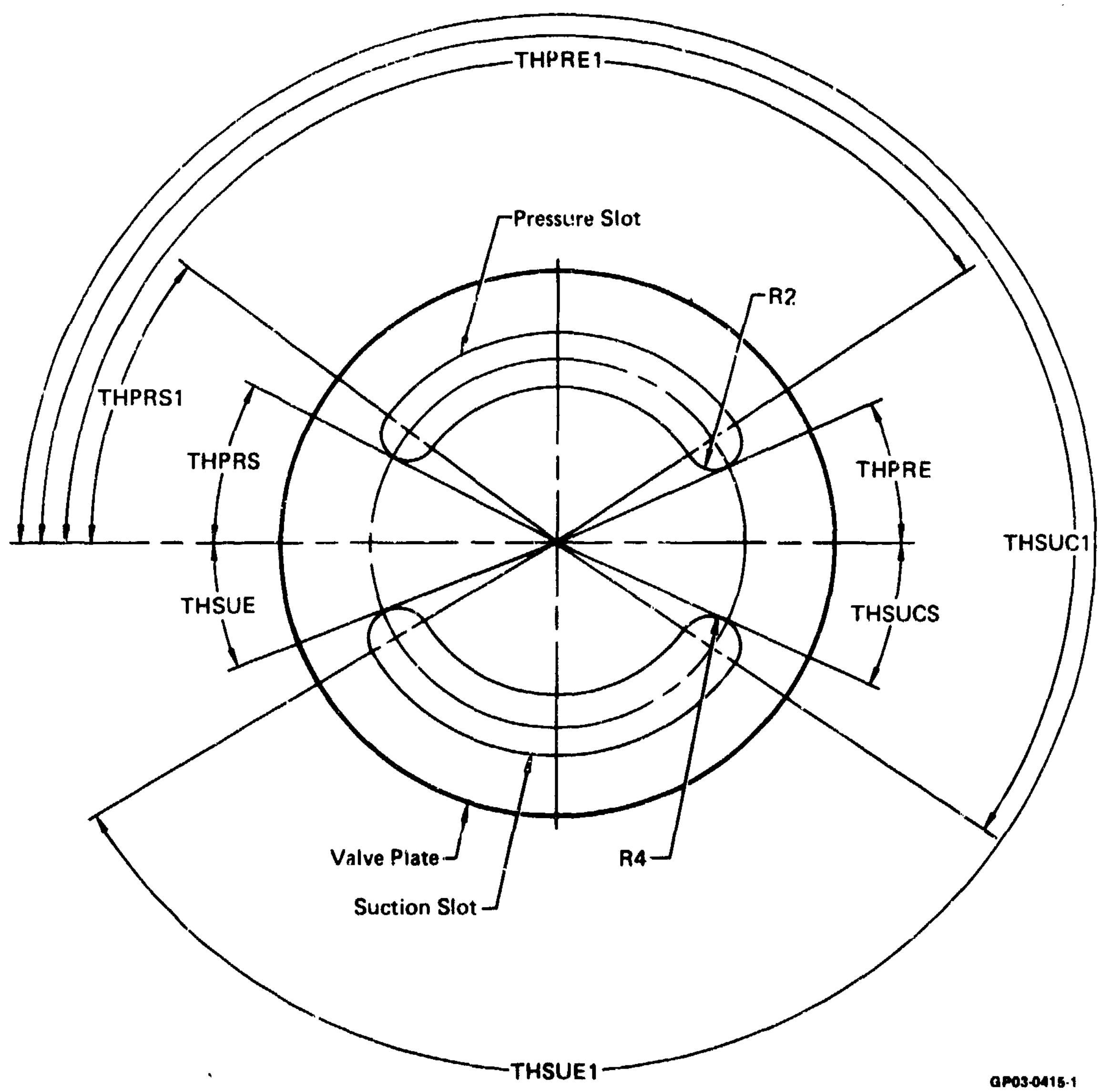


FIGURE 4-3
PUMP VALVE PLATE PARAMETERS

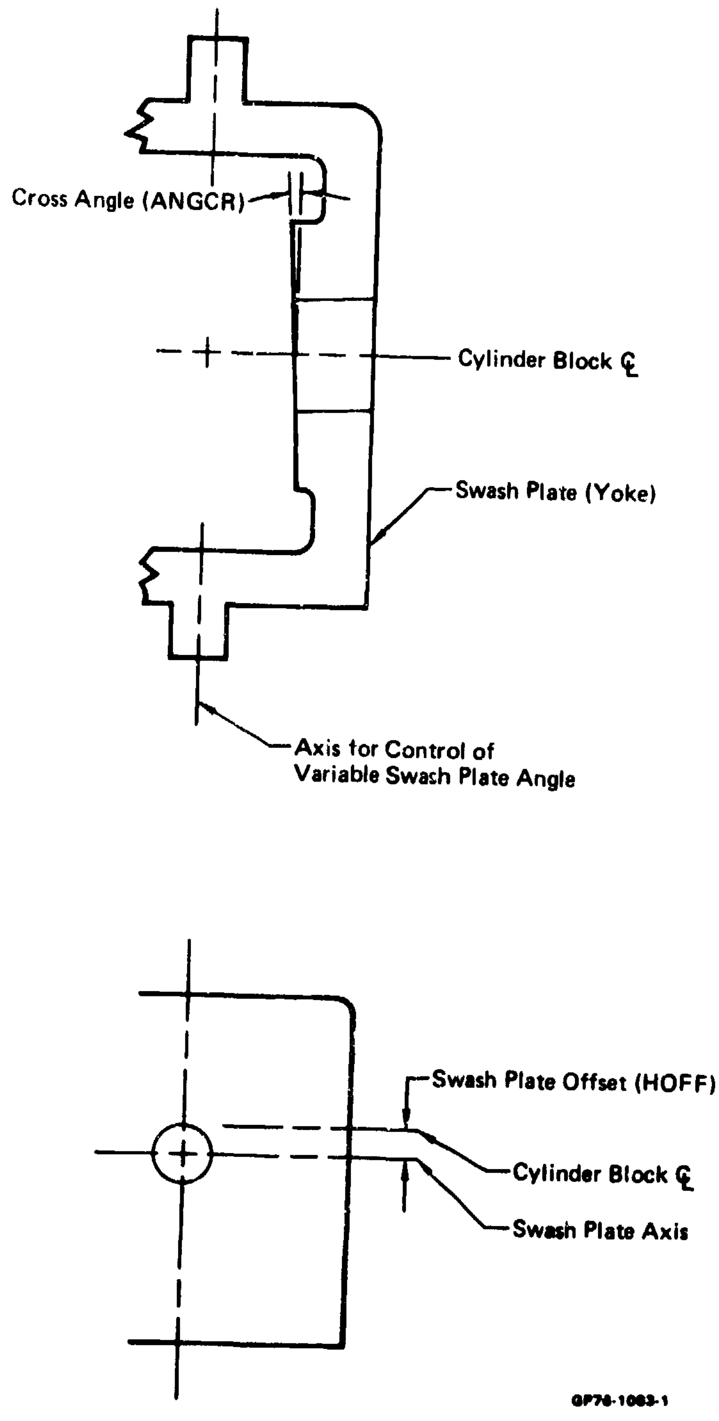
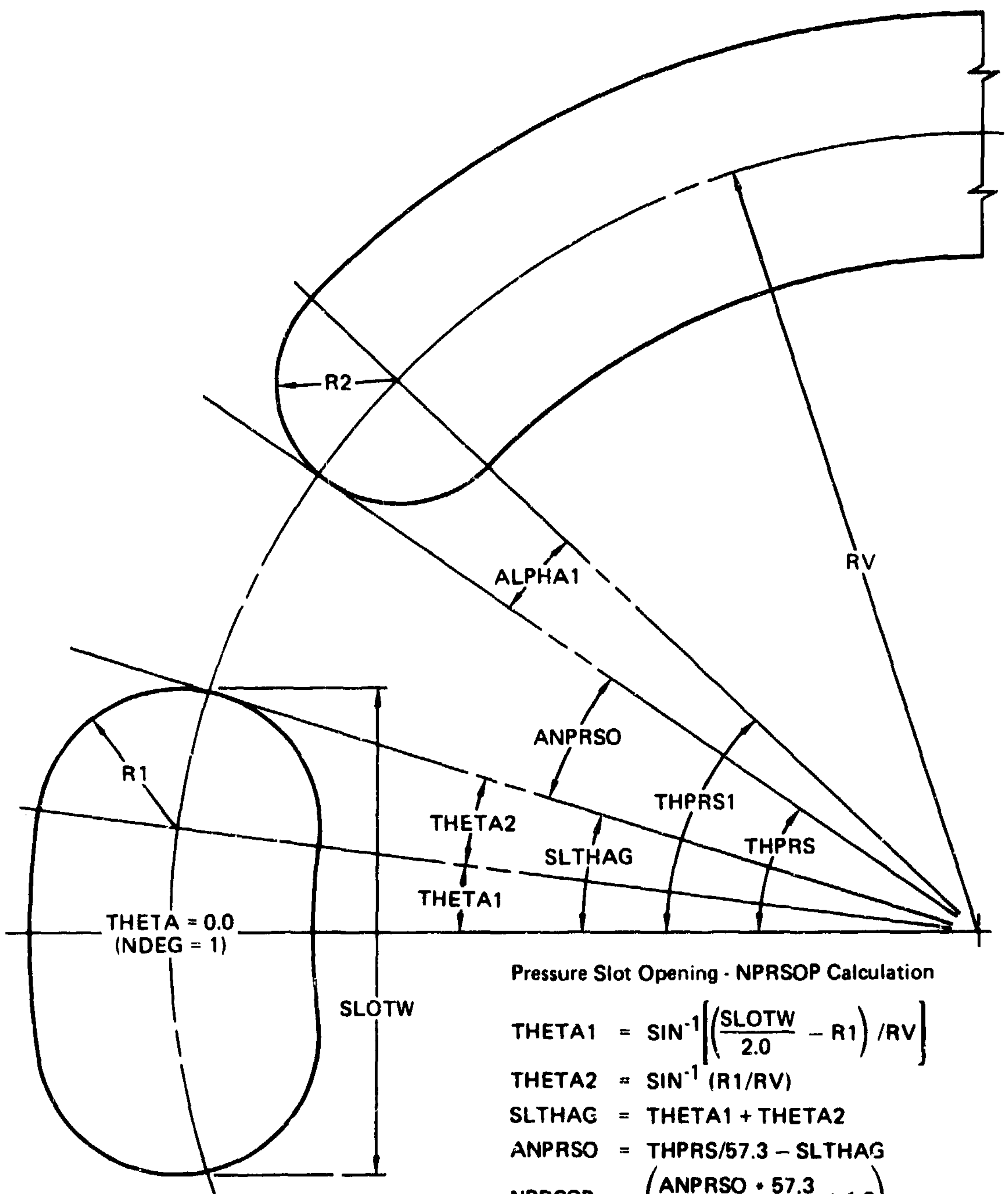


FIGURE 4-4
SWASH PLATE PARAMETERS



Pressure Slot Opening - NPRSOP Calculation

$$\text{THETA1} = \sin^{-1}\left[\left(\frac{\text{SLOTW}}{2.0} - \text{R1}\right)/\text{RV}\right]$$

$$\text{THETA2} = \sin^{-1}(\text{R1}/\text{RV})$$

$$\text{SLTHAG} = \text{THETA1} + \text{THETA2}$$

$$\text{ANPRSO} = \text{THPRS}/57.3 - \text{SLTHAG}$$

$$\text{NPRSOP} = \left(\frac{\text{ANPRSO} * 57.3}{\text{AINC}} + 1.0\right)$$

$$\text{ALPHA1} = \sin^{-1}(\text{R2}/\text{RV})$$

$$\text{THPRS1} = \text{THPRS}/57.3 + \text{ALPHA1}$$

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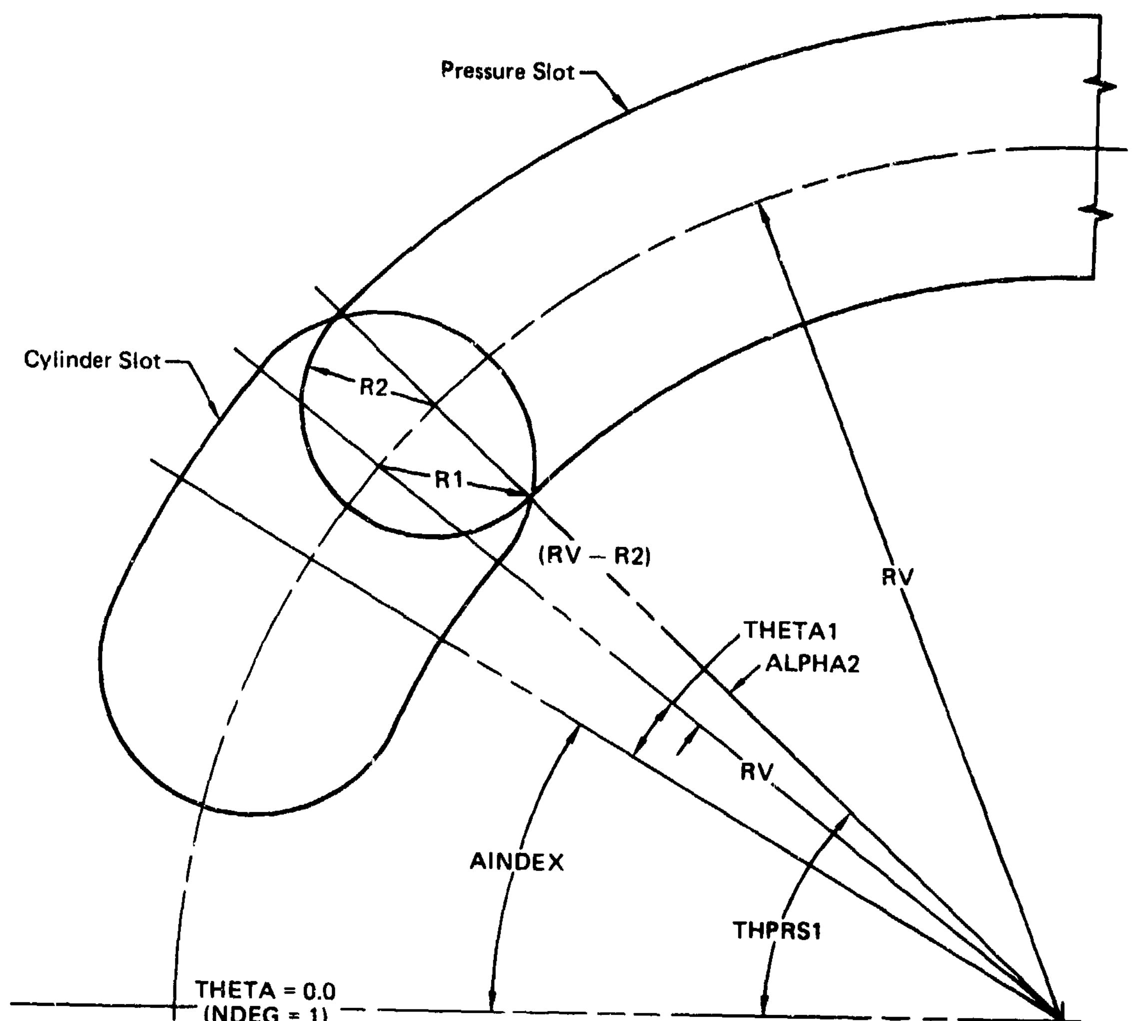
FIGURE 4-5
PUMP PORT VALVE AREA CALCULATIONS



Pressure Slot Opening - INDEX1 Calculation

$$ALPHA2 = \cos^{-1}\left[\frac{RV^2 + (RV - R2)^2 - R1^2}{2.0 \cdot RV \cdot (RV - R2)}\right]$$

$$AINDEX = THPRS1 - ALPH2A - THETA1$$

$$INDEX1 = \frac{AINDEX \cdot 57.3}{AINC} + 1.0$$

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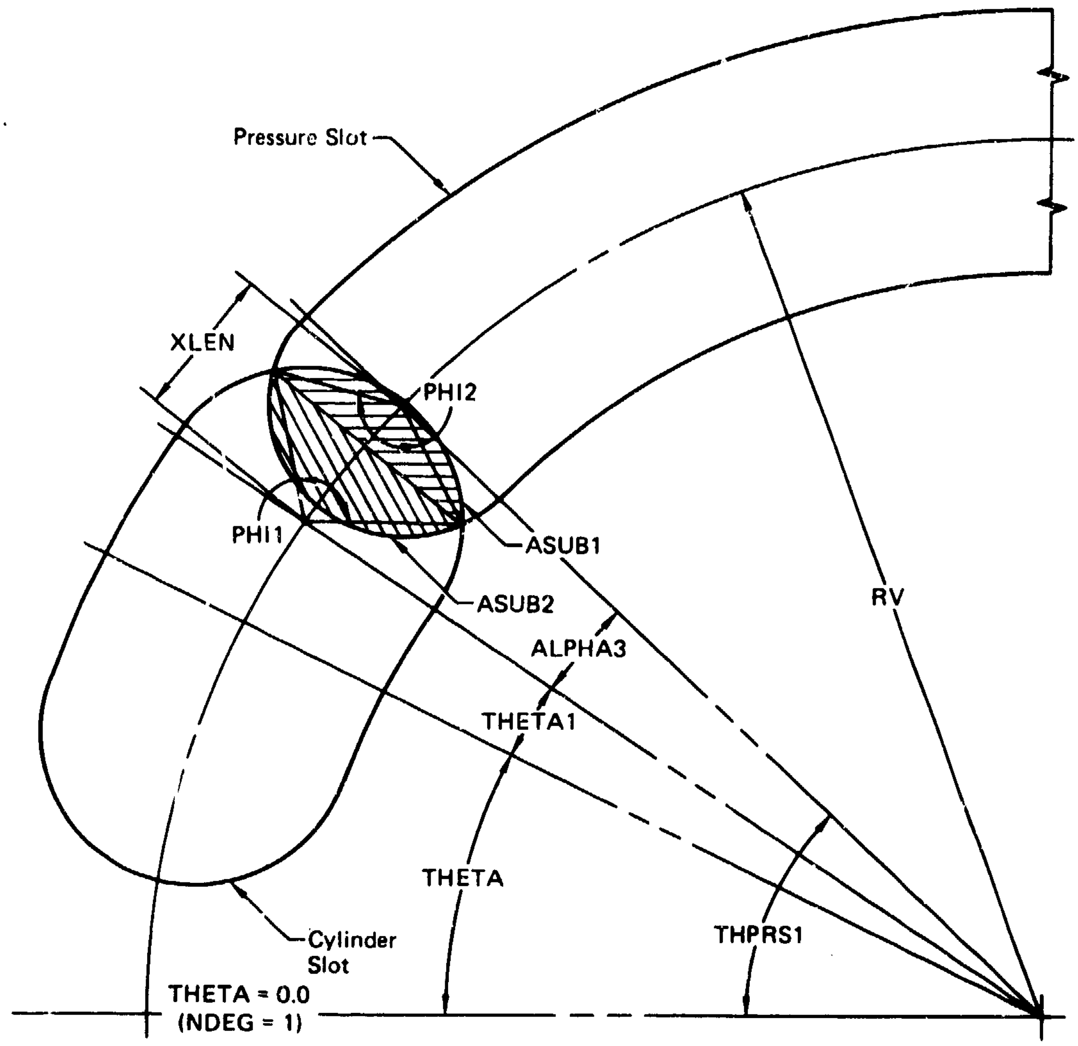
FIGURE 4-5 (Continued)
PUMP PORT VALVE AREA CALCULATIONS



Valve Area Calculation - NPRSOP + 1 $\leq$ NDEG $\leq$ INDEX1

$$\text{THETA} = \left(\frac{\text{NDEG}}{2.0} - 0.5\right)/57.3$$

$$\text{ALPHA3} = \text{THPRS1} - \text{THETA} - \text{THETA1}$$

$$\text{XLEN} = 2.0 * \text{RV} * \text{SIN}\left(\frac{\text{ALPHA3}}{2.0}\right)$$

$$\text{PHI1} = 2.0 * \text{COS}^{-1}\left(\frac{\text{R1}^2 + \text{XLEN}^2 - \text{R2}^2}{2.0 * \text{R1} * \text{XLEN}}\right)$$

$$\text{PHI2} = 2.0 * \text{COS}^{-1}\left(\frac{\text{R2}^2 + \text{XLEN}^2 - \text{R1}^2}{2.0 * \text{R1} * \text{XLEN}}\right)$$

$$\text{ASUB1} = \left(\frac{\text{R1}^2}{2.0}\right) * (\text{PHI1} - \text{SIN}(\text{PHI1})$$

$$\text{ASUB2} = \left(\frac{\text{R2}^2}{2.0}\right) * (\text{PHI2} - \text{SIN}(\text{PHI2})$$

$$\text{VAREA}(\text{NDEG}) = \text{ASUB1} + \text{ASUB2}$$

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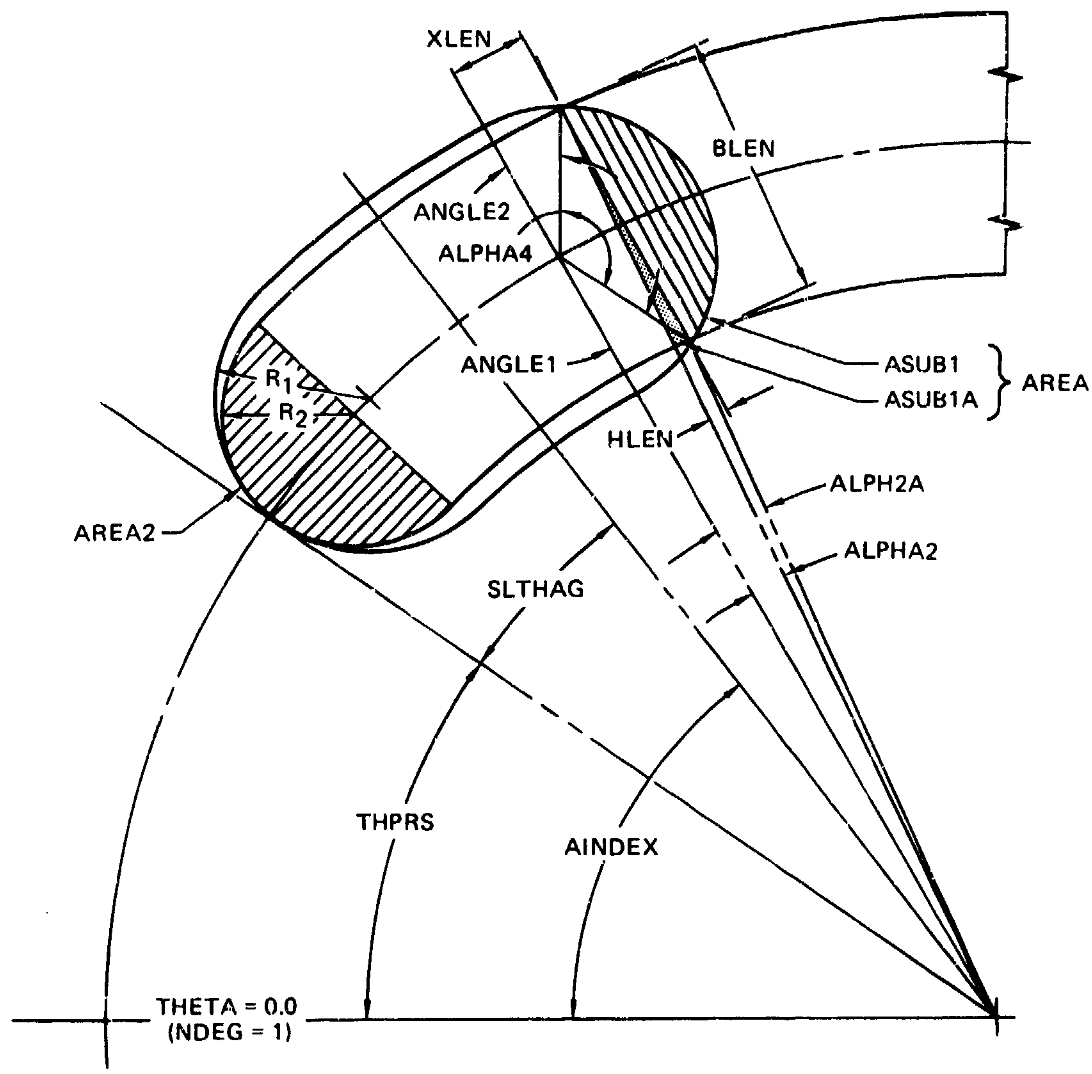
FIGURE 4-5 (Continued)
PUMP PORT VALVE AREA CALCULATIONS



Pressure Slot Opening - INDEX2 Calculation

$$ALPH2A = \cos^{-1}\left(\frac{R2^2 + (RV + R2)^2 - R1^2}{2.0 \cdot R2 \cdot (RV + R2)}\right)$$

$$BLEN = 2.0 \cdot R2$$

$$HLEN = 2.0 \cdot (RV - R2) \cdot SIN\left(\frac{ALPHA2 - ALPH2A}{2.0}\right)$$

$$ASUB1A = BLEN \cdot HLEN/2.0$$

$$ANGLE1 = \cos^{-1}\left(\frac{R1^2 + RV^2 - (RV - R2)^2}{2.0 \cdot R1 \cdot RV}\right)$$

$$XLEN = 2.0 \cdot (RV + R2) \cdot SIN\left(\frac{ALPH2A}{2.0}\right)$$

$$ANGLE2 = \cos^{-1}\left(\frac{R1^2 + R2^2 - XLEN^2}{2.0 \cdot R1 \cdot R2}\right)$$

$$ALPHA4 = \pi - ANGLE1 - ANGLE2$$

$$ASUB1 = \left(\frac{R1^2}{2.0}\right) \cdot (ALPHA4 - SIN\ (ALPHA4))$$

$$AREA1 = ASUB1 + ASUB1A$$

$$AREA2 = \pi \cdot R2^2/2.0$$

$$AINDEX = THPRS/57.3 + SLTHAG$$

$$INDEX2 = \frac{AINDEX \cdot 57.3}{AINC} + 1.0$$

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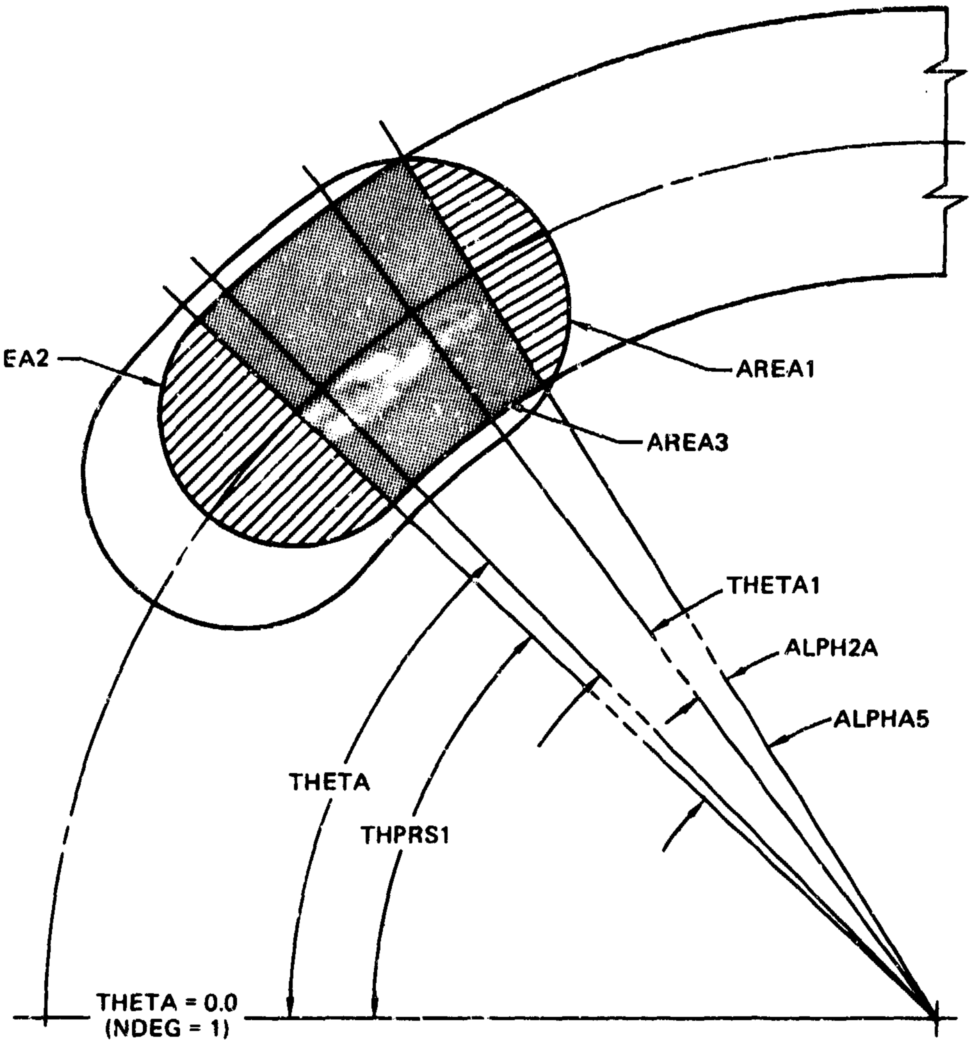
FIGURE 4-5 (Continued)
PUMP PORT VALVE AREA CALCULATIONS



Valve Area Calculation - INDEX1 + 1 $\leq$ NDEG $\leq$ INDEX2

$$\text{THETA} = \left(\frac{\text{NDEG}}{2.0} - 0.5\right) / 57.3$$

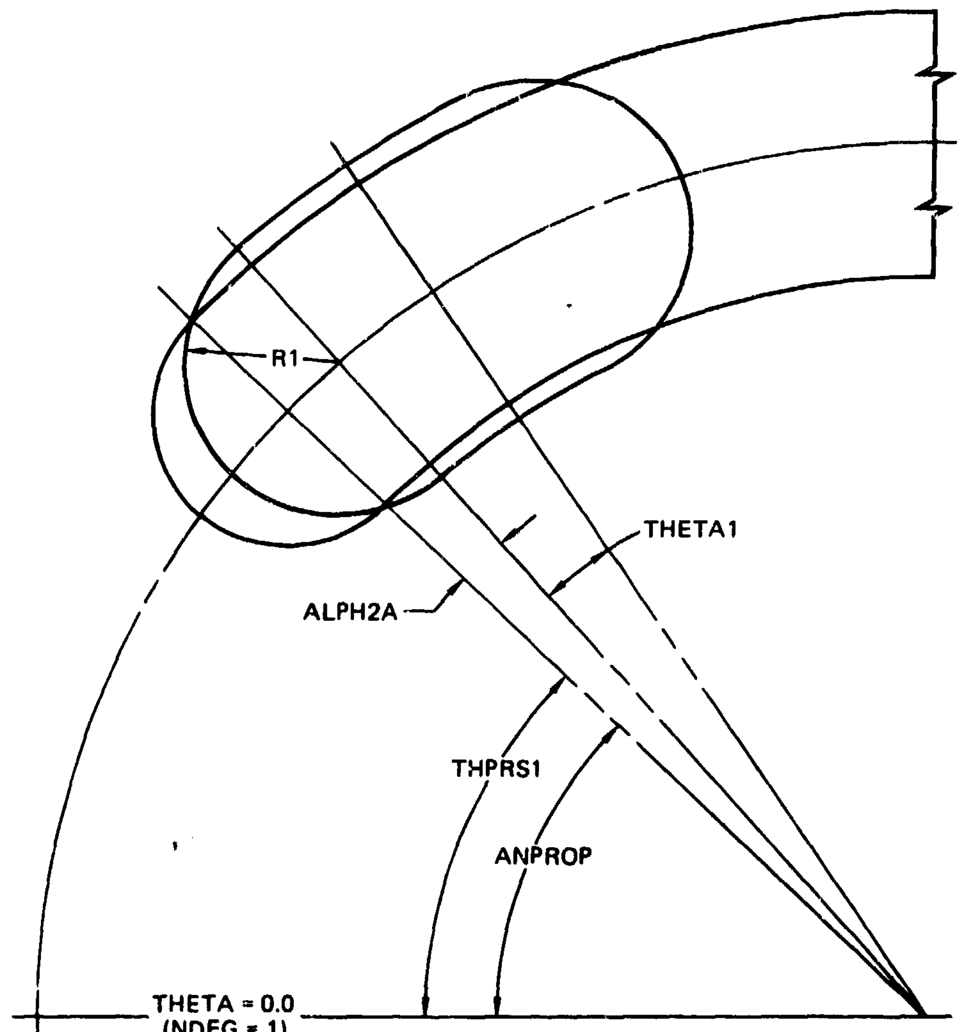
ALPHA5 = THETA1 + ALPH2A + (THETA − THRPS1)

AREA3 = 2.0 • RV • R2 • ALPHA5

VAREA (NDEG) = AREA1 + AREA2 + AREA3

GP03-0415-6

FIGURE 4-5 (Continued)
PUMP PORT VALVE AREA CALCULATIONS



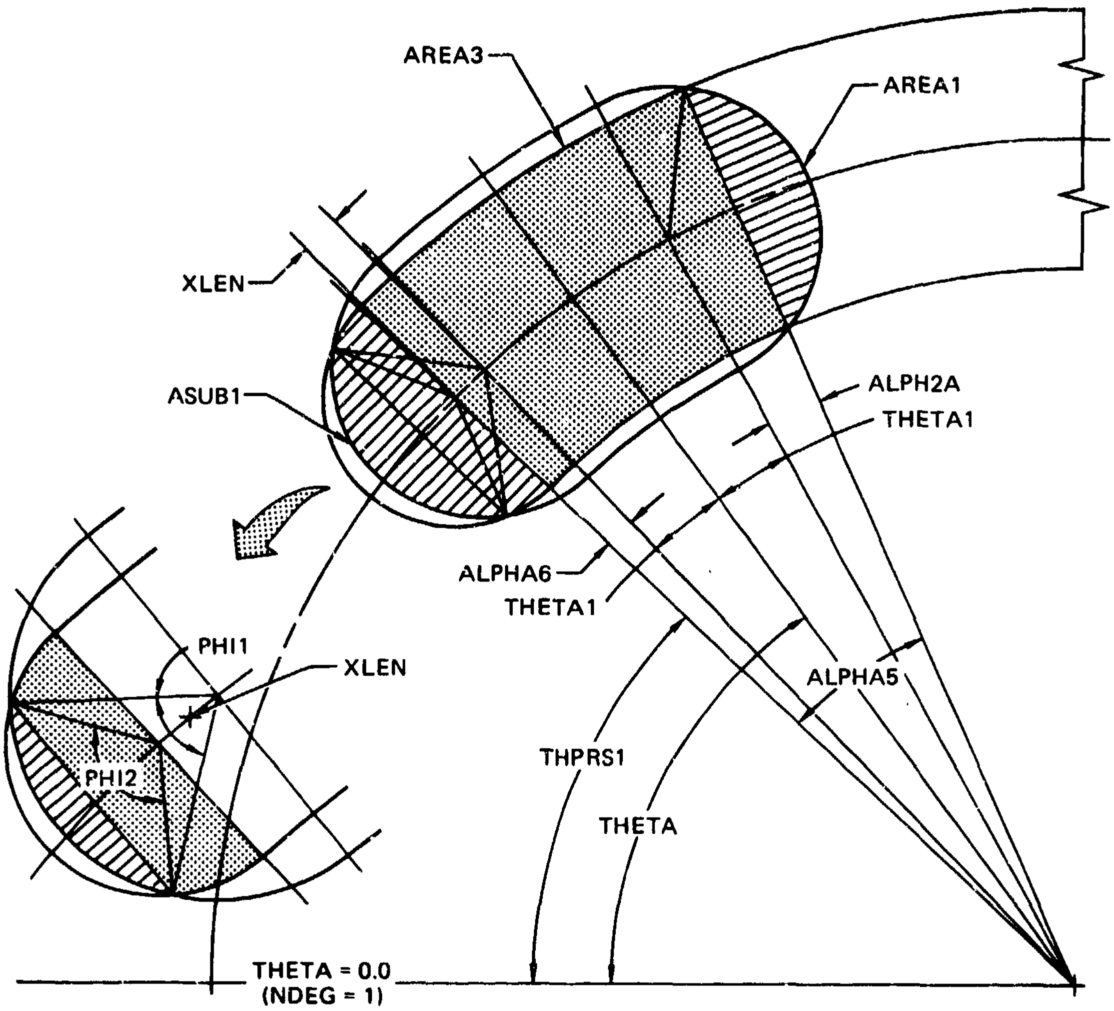
Pressure Slot Opening - NPROP Calculation

$$ANPROP = THPRS1 + ALPH2A + THETA1$$

$$NPROP = \frac{ANPROP * 57.3}{AINC} + 1.0$$

GP03-0416-7

FIGURE 4-5 (Continued)
PUMP PORT VALVE AREA CALCULATIONS



Valve Area Calculation - INDEX2 + 1 $\leq$ NDEG $\leq$ NPROP

$$\text{THETA} = \left(\frac{\text{NDEG}}{2.0} - 0.5\right) / 57.3$$

$$\text{ALPHA6} = \text{THETA} - \text{THETA1} - \text{THPRS1}$$

$$\text{ALPHA5} = 2.0 \cdot \text{THETA1} + \text{ALPHA6} + \text{ALPH2A}$$

$$\text{XLEN} = 2.0 \cdot \text{RV} \cdot \text{SIN}\left(\frac{\text{ALPHA6}}{2.0}\right)$$

$$\text{AREA3} = 2.0 \cdot \text{RV} \cdot \text{R2} \cdot \text{ALPHA5}$$

$$\text{ASUB1} = \frac{(\pi \cdot \text{R2}^2 - \text{R2}^2 \cdot [\text{PHI2} - \text{SIN}(\text{PHI2})] + \text{R1}^2 \cdot [\text{PHI1} - \text{SIN}(\text{PHI1})])}{2.0}$$

$$\text{PHI1} = 2.0 \cdot \text{COS}^{-1}\left(\frac{\text{R1}^2 + \text{XLEN}^2 - \text{R2}^2}{2.0 \cdot \text{R1} \cdot \text{XLEN}}\right)$$

$$\text{PHI2} = 2.0 \cdot \left[\pi - \text{COS}^{-1}\left(\frac{\text{R2}^2 + \text{XLEN}^2 - \text{R1}^2}{2.0 \cdot \text{R2} \cdot \text{XLEN}}\right)\right]$$

VAREA (NDEG) = AREA1 + ASUB1 + AREA3

GP03-0415-8

FIGURE 4-5 (Concluded)
PUMP PORT VALVE AREA CALCULATIONS

4.2.7 <u>Section 1 - Valve Area Calculation-Listing</u>

```
C
C     SECTION 1 VALVE AREA CALCULATION FOR FULL 360 DECREES REVOLUTION
C
      AINC=0.5
      ND9=360./AINC
      THETA1=(SLOTW/2.-R1)/RV
      THETA1=ASIN(THETA1)
      THETA2=R1/RV
      THETA2=ASIN(THETA2)
      SLTHAG=THETA1+THETA2
      NAPP=(180.+2.*SLTHAG*57.3-THPRS-THPRE)/360.*PISTNO+1.
      NAPS=(180.+2.*SLTHAG*57.3-THSUCS-THSUCE)/360.*PISTNO+1.
C
C     CALCULATE PRESSURE SLOT VALVE AREAS
C
      ALPHA1=R2/RV
      ALPHA1=ASIN(ALPHA1)
      THPRS1=THPRS/57.3+ALPHA1
      ANPRSO=THPRS/57.3-SLTHAG
      NPRSOP=ANPRSO*57.3/AINC+1.
      DO 400 NDEG=1,NPRSOP
      VAREA(NDEG)=0.0
  400 CONTINUE
      ALPHA2=(RV**2+(RV-R2)**2-R1**2)/(2.*RV*(RV-R2))
      IF(R2.GT.R1)ALPHA2=(RV**2+(RV-R1)**2-R2**2)/(2.*RV*(RV-R1))
      ALPHA2=ACOS(ALPHA2)
      AINDEX=THPRS1-ALPHA2-THETA1
      INDEX1=AINDEX*57.3/AINC+1.
      NBEG1=NPRSOP+1
      DO 405 NDEG=NBEG1,INDEX1
      THETA=(NDEG/2.-0.5)/57.3
      ALPHA3=(THPRS1-THETA-THETA1)/2.
      XLEN=2.*RV*SIN(ALPHA3)
      PHI1=(R1**2+XLEN**2-R2**2)/(2.*R1*XLEN)
      PHI1=2.*ACOS(PHI1)
      PHI2=(R2**2+XLEN**2-R1**2)/(2.*R2*XLEN)
      PHI2=2.*ACOS(PHI2)
      ASUB1=(R1**2)*(PHI1-SIN(PHI1))/2.
      ASUB2=(R2**2)*(PHI2-SIN(PHI2))/2.
      VAREA(NDEG)=ASUB1+ASUB2
  405 CONTINUE
      ALPH2A=(RV**2+(RV+R2)**2-R1**2)/(2.*RV*(RV+R2))
      IF(R2.GT.R1)ALPH2A=(RV**2+(RV+R1)**2-R2**2)/(2.*RV*(RV+R1))
      ALPH2A=ACOS(ALPH2A)
      BLEN=2.*R2
      IF(R2.GT.R1)BLEN=2.*R1
      ANLEN=(ALPHA2-ALPH2A)/2.
      HLEN=2.*(RV-R2)*SIN(ANLEN)
      IF(R2.GT.R1)HLEN=2.*(RV-R1)*SIN(ANLEN)
      ASUB1A=BLEN*HLEN/2.
```

4.2.7 (Continued)

```
      ANGLE1=(R1**2+RV**2-(RV-R2)**2)/(2.*R1*RV)
      IF(R2.GT.R1)ANGLE1=(R2**2+RV**2-(RV-R1)**2)/(2.*R2*RV)
      ANGLE1=ACOS(ANGLE1)
      AXLEN=ALPH2A/2.
      XLEN=2.*(RV+R2)*SIN(AXLEN)
      IF(R2.GT.R1)XLEN=2.*(RV+R1)*SIN(AXLEN)
      ANGLE2=(R1**2+R2**2-XLEN**2)/(2.*R1*R2)
      ANGLE2=ACOS(ANGLE2)
      ALPHA4=PI-ANGLE1-ANGLE2
      ASUB1=(R1**2)*(ALPHA4-SIN(ALPHA4))/2.
      IF(R2.GT.R1)ASUB1=(R2**2)*(ALPHA4-SIN(ALPHA4))/2.
      AREA1=ASUB1+ASUB1A
      AREA2=PI*R2**2/2.
      IF(R2.GT.R1)AREA2=PI*R1**2/2.
      AINDEX=THPRS/57.3+SLTHAG
      IF(R2.GT.R1)AINDEX=THPRS1-ALPH2A+THETA1
      INDEX2=AINDEX*57.3/AINC+1.
      NBEG2=INDEX1+1
      DO 410 NDEG=NBEG2,INDEX2
      THETA=(NDEG/2.-0.5)/57.3
      ALPHA5=THETA1+THETA-THPRS1+ALPH2A
      AREA3=2.*RV*R2*ALPHA5
      IF(R2.GT.R1)AREA3=2.*RV*R1*ALPHA5
      VAREA(NDEG)=AREA1+AREA2+AREA3
  410 CONTINUE
      ANPROP=THPRS1+ALPH2A+THETA1
      IF(R2.GT.R1)ANPROP=THPRS/57.3+SLTHAG
      NPROP=ANPROP*57.3/AINC+1.
      NBEG3=INDEX2+1
      DO 415 NDEG=NBEG3,NPROP
      THETA=(NDEG/2.-0.5)/57.3
      ALPHA6=(THETA-THETA1-THPRS1)/2.
      IF(R2.GT.R1)ALPHA6=(THPRS1-THETA+THETA1)/2.
      ALPHA5=2.*(THETA1+ALPHA6)+ALPH2A
      XLEN=2.*RV*SIN(ALPHA6)
      PHI1=(R1**2+XLEN**2-R2**2)/(2.*R1*XLEN)
      PHI2=(R2**2+XLEN**2-R1**2)/(2.*R2*XLEN)
      PHI1=2.*ACOS(PHI1)
      IF(R2.GT.R1)PHI1=2.*(PI-ACOS(PHI1))
      PHI2=2.*(PI-ACOS(PHI2))
      IF(R2.GT.R1)PHI2=2.*ACOS(PHI2)
      ASUB1=(PI*(R2**2)-(R2**2)*(PHI2-SIN(PHI2))+
     1(R1**2)*(PHI1-SIN(PHI1)))/2.
      IF(R2.GT.R1)ASUB1=(PI*(R1**2)-(R1**2)*(PHI1-SIN(PHI1))+
     1(R2**2)*(PHI2-SIN(PHI2)))/2.
      AREA3=2.*RV*R2*ALPHA5
      IF(R2.GT.R1)AREA3=4.*RV*R1*THETA1
      VAREA(NDEG)=AREA1+ASUB1+AREA3
      IF(R2.GT.R1)VAREA(NDEG)=ASUB1+AREA2+AREA3
  415 CONTINUE
```

4.2.7 (Continued)

```
      THPRE1=PI-THPRE/57.3-ALPHA1
      ANPRSC=THPRE1-THETA1-ALPH2A
      IF(R2.GT.R1)ANPRSC=PI-THPRE/57.3-SLTHAG
      NPRSCL=ANPRSC*57.3/AINC+1.
      NBEG4=NPROP+1
      DO 420 NDEG=NBEG4,NPRSCL
      VAREA(NDEG)=2.*AREA1+4.*RV*R2*(THETA1+ALPH2A)
      IF(R2.GT.R1)VAREA(NDEG)=2.*AREA2+AREA3
  420 CONTINUE
      AINDEX=PI-THPRE/57.3-SLTHAG
      IF(R2.GT.R1)AINDEX=THPRE1+ALPH2A -THETA1
      INDEX3=AINDEX*57.3/AINC+1.
      NBEG5=NPRSCL+1
      DO 425 NDEG=NBEG5,INDEX3
      THETA=(NDEG/2.-0.5)/57.3
      ALPHA6=(THPRE1-THETA-THETA1)/2.
      IF(R2.GT.R1)ALPHA6=(THETA+THETA1-THPRE1)/2.
      ALPHA5=2.*(ALPHA6+THETA1)+ALPH2A
      XLEN=2.*RV*SIN(ALPHA6)
      PHI1=(R1**2+XLEN**2-R2**2)/(2.*R1*XLEN)
      PHI2=(R2**2+XLEN**2-R1**2)/(2.*R2*XLEN)
      PHI1=2.*ACOS(PHI1)
      IF(R2.GT.R1)PHI1=2.*(PI-ACOS(PHI1))
      PHI2=2.*(PI-ACOS(PHI2))
      IF(R2.GT.R1)PHI2=2.*ACOS(PHI2)
      ASUB1=(PI*(R2**2)-(R2**2)*(PHI2-SIN(PHI2))+
     1(R1**2)*(PHI1-SIN(PHI1)))/2.
      IF(R2.GT.R1)ASUB1=(PI*(R1**2)-(R1**2)*(PHI1-SIN(PHI1))+
     1(R2**2)*(PHI2-SIN(PHI2)))/2.
      AREA3=2.*RV*R2*ALPHA5
      IF(R2.GT.R1)AREA3=4.*RV*R1*THETA1
      VAREA(NDEG)=ASUB1+AREA1+AREA3
      IF(R2.GT.R1)VAREA(NDEG)=ASUB1+AREA2+AREA3
  425 CONTINUE
      AINDEX=THPRE1+ALPHA2+THETA1
      INDEX4=AINDEX*57.3/AINC+1.
      NBEG6=INDEX3+1
      DO 430 NDEG=NBEG6,INDEX4
      THETA=(NDEG/2.-0.5)/57.3
      ALPHA5=THPRE1-THETA+THETA1+ALPH2A
      AREA3=2.*RV*R2*ALPHA5
      IF(R2.GT.R1)AREA3=2.*RV*R1*ALPHA5
      VAREA(NDEG)=AREA1+AREA2+AREA3
  430 CONTINUE
      ANPRCL=PI-THPRE/57.3+SLTHAG
      NPRCL=ANPRCL*57.3/AINC+1.
      NBEG7=INDEX4+1
      DO 435 NDEG=NBEG7,NPRCL
      THETA=(NDEG/2.-0.5)/57.3
      ALPHA3=(THETA-THETA1-THPRE1)/2.
```

4.2.7 (Continued)

```
      XLEN=2.*RV*SIN(ALPHA3)
      PHI1=(R1**2+XLEN**2-R2**2)/(2.*R1*XLEN)
      PHI1=2.*ACOS(PHI1)
      PHI2=(R2**2+XLEN**2-R1**2)/(2.*R2*XLEN)
      PHI2=2.*ACOS(PHI2)
      ASUB1=(R1**2)*(PHI1-SIN(PHI1))/2.
      ASUB2=(R2**2)*(PHI2-SIN(PHI2))/2.
      VAREA(NDEG)=ASUB1+ASUB2
  435 CONTINUE
C
C     CALCULATE SUCTION SLOT VALVE AREAS
C
      ALPHA1=R4/RV
      ALPHA1=ASIN(ALPHA1)
      THSUC1=PI+THSUCS/57.3+ALPHA1
      ANSUSO=PI+THSUCS/57.3-SLTMAG
      NSUSOP=ANSUSO*57.3/AINC+1.
      NBEG1=NPRCL+1
      DO 440 NDEG=NBEG1,NSUSOP
      VAREA(NDEG)=0.0
  440 CONTINUE
      ALPHA2=(RV**2+(RV-R1)**2-R4**2)/(2.*RV*(RV-R1))
      IF(R1.GT.R4)ALPHA2=(RV**2+(RV-R4)**2-R1**2)/(2.*RV*(RV-R4))
      ALPHA2=ACOS(ALPHA2)
      AINDEX=THSUC1-ALPHA2-THETA1
      INDEX1=AINDEX*57.3/AINC+1.
      NBEG1=NSUSOP+1
      DO 445 NDEG=NBEG1,INDEX1
      THETA=(NDEG/2.-0.5)/57.3
      ALPHA3=(THSUC1-THETA-THETA1)/2.
      XLEN=2.*RV*SIN(ALPHA3)
      PHI1=(R1**2+XLEN**2-R4**2)/(2.*R1*XLEN)
      PHI1=2.*ACOS(PHI1)
      PHI4=(R4**2+XLEN**2-R1**2)/(2.*R4*XLEN)
      PHI4=2.*ACOS(PHI4)
      ASUB1=(R1**2)*(PHI1-SIN(PHI1))/2.
      ASUB2=(R4**2)*(PHI4-SIN(PHI4))/2.
      VAREA(NDEG)=ASUB1+ASUB2
  445 CONTINUE
      ALPH2A=(RV**2+(RV+R1)**2-R4**2)/(2.*RV*(RV+R1))
      IF(R1.GT.R4)ALPH2A=(RV**2+(RV+R4)**2-R1**2)/(2.*RV*(RV+R4))
      ALPH2A=ACOS(ALPH2A)
      BLEN=2.*R1
      IF(R1.GT.R4)BLEN=2.*R4
      AHLEN=(ALPHA2-ALPH2A)/2.
      HLEN=2.*(RV-R1)*SIN(AHLEN)
      IF(R1.GT.R4)HLEN=2.*(RV-R4)*SIN(AHLEN)
      ASUB1A=BLEN*HLEN/2.
      ANGLE1=(RV**2+R4**2-(RV-R1)**2)/(2.*RV*R4)
      IF(R1.GT.R4)ANGLE1=(RV**2+R1**2-(RV-R4)**2)/(2.*RV*R1)
```

4.2.7 (Continued)

```
      ANGLE1=ACOS(ANGLE1)
      XLEN=SQRT(2.*(RV+R4)**2-2.*(RV+R4)**2*COS(ALPH2A))
      IF(R1.GT.R4)XLEN=SQRT(2.*(RV+R1)**2-2.*(RV+R1)**2*COS(ALPH2A))
      ANGLE2=(2.*R4**2-XLEN**2)/(2.*R4**2)
      IF(R1.GT.R4)ANGLE2=(2.*R1**2-XLEN**2)/(2.*R1**2)
      ANGLE2=ACOS(ANGLE2)
      ALPHA4=PI-ANGLE1-ANGLE2
      ASUB1=(R4**2)*(ALPHA4-SIN(ALPHA4))/2.
      IF(R1.GT.R4)ASUB1=(R1**2)*(ALPHA4-SIN(ALPHA4))/2.
      AREA1=ASUB1+ASUB1A
      AREA2=PI*(R1**2)/2.
      IF(R1.GT.R4)AREA2=PI*R4**2/2.
      AINDEX=THSUC1-ALPH2A+THETA1
      IF(R1.GT.R4)AINDEX=THSUC1-ALPHA1+SLTHAG
      INDEX2=AINDEX*57.3/AINC+1.
      NBEG2=INDEX1+1
      DO 450 NDEG=NBEG2,INDEX2
      THETA=(NDEG/2.-0.5)/57.3
      ALPHA5=THETA+THETA1+ALPH2A-THSUC1
      AREA3=2.*RV*R1*ALPHA5
      IF(R1.GT.R4)AREA3=2.*RV*R4*ALPHA5
      VAREA(NDEG)=AREA1+AREA2+AREA3
  450 CONTINUE
      ANSUOP=THSUC1-ALPHA1+SLTHAG
      IF(R1.GT.R4)ANSUOP=THSUC1+ALPH2A+THETA1
      NSUOP=ANSUOP*57.3/AINC+1.
      NBEG3=INDEX2+1
      DO 455 NDEG=NBEG3,NSUOP
      THETA=(NDEG/2.-0.5)/57.3
      ALPHA6=(THSUC1-THETA+THETA1)/2.
      IF(R1.GT.R4)ALPHA6=(THETA-THETA1-THSUC1)/2.
      ALPHA5=2.*(THETA1+ALPHA6)+ALPH2A
      XLEN=2.*RV*SIN(ALPHA6)
      PHI1=(R1**2+XLEN**2-R4**2)/(2.*R1*XLEN)
      PHI4=(R4**2+XLEN**2-R1**2)/(2.*R4*XLEN)
      PHI1=2.*(PI-ACOS(PHI1))
      IF(R1.GT.R4)PHI1=2.*ACOS(PHI1)
      PHI4=2.*ACOS(PHI4)
      IF(R1.GT.R4)PHI4=2.*(PI-ACOS(PHI4))
      ASUB1=(PI*(R1**2)-(R1**2)*(PHI1-SIN(PHI1))+
     1(R4**2)*(PHI4-SIN(PHI4)))/2.
      IF(R1.GT.R4)ASUB1=(PI*(R4**2)-(R4**2)*(PHI4-SIN(PHI4))+
     1(R1**2)*(PHI1-SIN(PHI1)))/2.
      AREA3=4.*RV*R1*THETA1
      IF(R1.GT.R4)AREA3=2.*RV*R4*ALPHA5
      VAREA(NDEG)=ASUB1+AREA2+AREA3
      IF(R1.GT.R4)VAREA(NDEG)=AREA1+ASUB1+AREA3
  455 CONTINUE
      THSUE1=2.*PI-THSUCE/57.3-ALPHA1
      ANSUSC=2.*PI-THSUCE/57.3-SLTHAG
```

4.2.7 (Continued)

```
      IF(R1.GT.R4)ANSUSC=THSUE1-ALPH2A-THETA1
      NSUSCL=ANSUSC*57.3/AINC+1.
      NBEG4=NSUOP+1
      DO 460 NDEG=NBEG4,NSUSCL
      VAREA(NDEG)=2.*AREA2+4.*RV*R1*THETA1
      IF(R1.GT.R4)VAREA(NDEG)=2.*AREA1+4.*RV*R4*(THETA1+ALPH2A)
  460 CONTINUE
      AINDEX=THSUE1+ALPH2A-THETA1
      IF(R1.GT.R4)AINDEX=2.*PI-THSUCE/57.3-SLTHAG
      INDEX3=AINDEX*57.3/AINC+1.
      NBEG5=NSUSCL+1
      DO 465 NDEG=NBEG5,INDEX3
      THETA=(NDEG/2.-0.5)/57.3
      ALPHA6=(THETA+THETA1-THSUE1)/2.
      IF(R1.GT.R4)ALPHA6=(THSUE1-THETA-THETA1)/2.
      ALPHA5=2.*(THETA1+ALPHA6)+ALPH2A
      XLEN=2.*RV*SIN(ALPHA6)
      PHI1=(R1**2+XLEN**2-R4**2)/(2.*R1*XLEN)
      PHI4=(R4**2+XLEN**2-R1**2)/(2.*R4*XLEN)
      PHI1=2.*(PI-ACOS(PHI1))
      IF(R1.GT.R4)PHI1=2.*ACOS(PHI1)
      PHI4=2.*ACOS(PHI4)
      IF(R1.GT.R4)PHI4=2.*(PI-ACOS(PHI4))
      ASUB1=(PI*(R1**2)-(R1**2)*(PHI1-SIN(PHI1))+
     1(R4**2)*(PHI4-SIN(PHI4)))/2.
      IF(R1.GT.R4)ASUB1=(PI*(R4**2)-(R4**2)*(PHI4-SIN(PHI4))+
     1(R1**2)*(PHI1-SIN(PHI1)))/2.
      AREA3=4.*RV*R1*THETA1
      IF(R1.GT.R4)AREA3=2.*RV*R4*ALPHA5
      VAREA(NDEG)=ASUB1+AREA2+AREA3
      IF(R1.GT.R4)VAREA(NDEG)=AREA1+ASUB1+AREA3
  465 CONTINUE
      AINDEX=THSUE1+THETA1+ALPH2A
      INDEX4=AINDEX*57.3/AINC+1.
      NBEG6=INDEX3+1
      DO 470 NDEG=NBEG6,INDEX4
      THETA=(NDEG/2.-0.5)/57.3
      ALPHA5=THSUE1+ALPH2A-THETA+THETA1
      AREA3=2.*RV*R1*ALPHA5
      IF(R1.GT.R4)AREA3=2.*RV*R4*ALPHA5
      VAREA(NDEG)=AREA1+AREA2+AREA3
  470 CONTINUE
      ANSUCL=2.*PI-THSUCE/57.3+SLTHAG
      NSUCL=ANSUCL*57.3/AINC+1.
      NBEG7=INDEX4+1
      DO 475 NDEG=NBEG7,NSUCL
      THETA=(NDEG/2.-0.5)/57.3
      ALPHA3=(THETA-THETA1-THSUE1)/2.
      XLEN=2.*RV*SIN(ALPHA3)
      PHI1=(R1**2+XLEN**2-R4**2)/(2.*R1*XLEN)
```

4.2.7 (Continued)

```
      PHI4=(R4**2+XLEN**2-R1**2)/(2.*R4*XLEN)
      PHI1=2.*ACOS(PHI1)
      PHI4=2.*ACOS(PHI4)
      ASUB1=(R1**2)*(PHI1-SIN(PHI1))/2.
      ASUB2=(R4**2)*(PHI4-SIN(PHI4))/2.
      VAREA(NDEG)=ASUB1+ASUB2
  475 CONTINUE
      NBEG8=NSUCL+1
      DO 480 NDEG=NBEG8,721
      VAREA(NDEG)=0.0
  480 CONTINUE
      WRITE(6,490)NPRSOP,NSUSOP,NPROP,NSUOP,NPRSCL,NSUSCL,NPRCL,NSUCL
  490 FORMAT(//,7X,'*** VALVE AREA INDEXES ***',//,5X,'NPRSOP =',I7,5X,
     1'NSUSOP =',I7,/,5X,'NPROP  =',I7,5X,'NSUOP  =',I7,/,5X,'NPRSCL =',
     2I7,5X,'NSUSCL =',I7,/,5X,'NPRCL  =',I7,5X,'NSUCL  =',I7,//)
      IF(IOPT(1).NE.1)GOTO 510
      WRITE(6,495)
  495 FORMAT('1',5X,'  NDEG  ',5X,'THETA',5X,'VAREA(NDEG)',//)
      DO 505 NDEG=1,721
      THETA=(NDEG/2.-0.5)/57.3
      WRITE(6,500)NDEG,THETA*57.3,VAREA(NDEG)
  500 FORMAT(8X,I3,5X,F10.6,2X,F10.6)
  505 CONTINUE
  510 CONTINUE
      R1SQ=(RV+R2)**2
      R2SQ=(RV-R2)**2
      ANGLE=(180.-THPRS-THPRE)/57.3
      AREA=.5*ANGLE*(R1SQ-R2SQ)
      DIAM=SQRT(AREA*4./PI)
```

4.3 SECTION 2 - STEADY STATE SWASH ANGLE AND OUTPUT PRESSURE CALCULATION

Section 2 calculates the steady state swash angle as a function of pump speed and circuit overboard flow. If the pump is saturated, i.e., overboard flow (QOVB) is greater than the capacity of the pump at a given RPM, the steady state outlet pressure at which the pump can supply the demand is calculated.

The flow constants QWBEG, QWEND, QZERO, and QMAXI are calculated on the first call of PUMP. These constants define a flow capability "envelope" which is used to estimate the steady state swash angle (ASWASH) or outlet pressure (HPRESS) for the first and each succeeding RPM. Calculation of the flow constants is bypassed on subsequent calls of PUMP.

4.3.1 Math Model

4.3.1.1 Flow Capability Envelope - Upon entering Section 2 on the first call of PUMP, the swash angle ASWASH is set to the input maximum SWASH and steady state outlet pressure HPRESS is set equal to input steady state outlet pressure PRESS. QWBEG is set equal to the calculated steady state flow QQFC(1) that results from Fourier Analysis of the values obtained in Section 4 at this condition of ASWASH and HPRESS with speed Y equal to the starting speed WSTART (see Sections 4.5 and 4.6). Speed is then set to the maximum input speed WEND. Flow calculations are repeated and QWEND is set equal to the resulting QQFC(1) from Fourier Analysis. QZERO is the value that results from straight line extrapolation of QWEND and QWBEG to zero RPM.

HPRESS is reduced to zero and the flow calculations are again repeated with speed and swash angle at their respective maximums. QMAXI is set equal to the resulting QQFC(1) from Fourier Analysis. These flow constants define an "envelope" of pump steady state flow capability as shown in Figure 4-6.

4.3.1.2 Swash Angle Calculation - Pump speed Y is set back to the starting RPM WSTART and the program is ready to estimate the swash angle required to provide

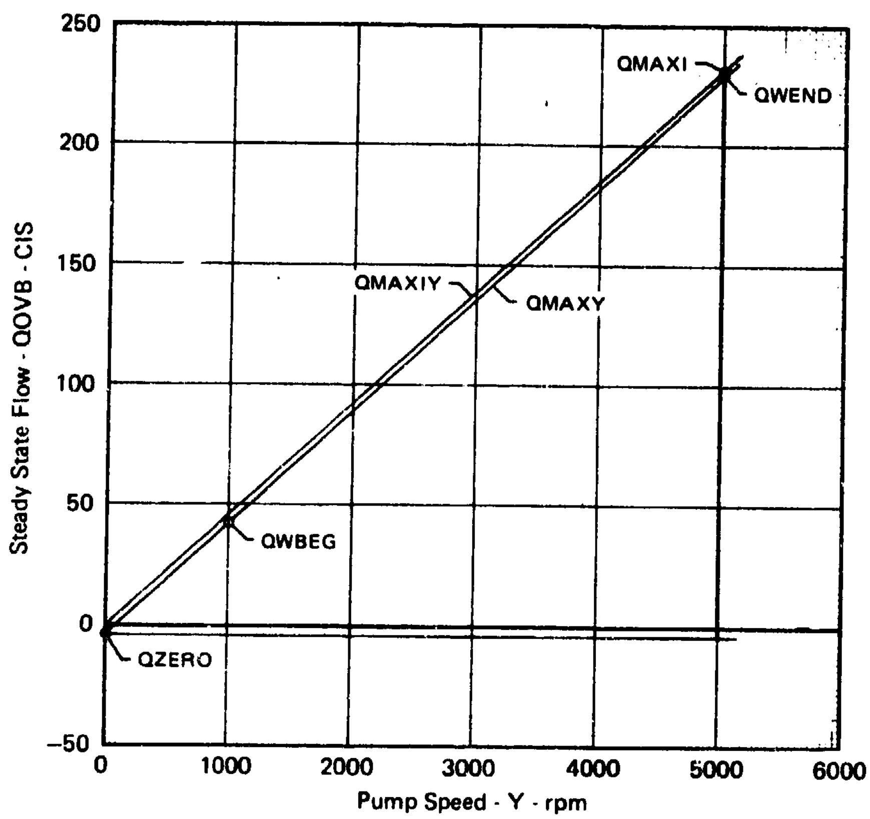


FIGURE 4-6
PUMP FLOW CAPABILITY ENVELOPE

the input steady state overboard flow QOVB. For the first speed (and for each succeeding RPM), the maximum swash angle/maximum pressure (QMAXY) and maximum swash angle/zero pressure (QMAXIY) are determined.

If QOVB is less than QMAXIY, the swash angle is estimated by linear interpolation between QZERO and QMAXY as shown in equation (1).

$$\text{ASWASH} = \frac{\text{SWASH}}{57.3} - \frac{\text{SWASH}}{57.3} * \frac{(\text{QMAXY-QOVB})}{(\text{QMAXY-QZERO})} \tag{1}$$

If ASWASH is less than SWASH, flow values for this condition are obtained and the resulting calculated steady state flow QQFC(1) is compared to QOVB (see Section 4.6). If the flow error FLERR(IFL) is greater than 0.01 CIS, ASWASH is adjusted by an incremental swash angle DSWERR, which is the calculated swash angle required to provide a flow increment equal to the flow error. DSWERR is also obtained by linear interpolation as shown in equation (2).

$$\text{DSWERR} = \frac{\text{SWASH}}{57.3} \quad \frac{\text{SWASH}}{57.3} \quad \frac{(\text{QMAXY-(QZERO+ABS(FLERR(IFL)))}}{(\text{QMAXY-QZERO})} \tag{2}$$

The process of determining flow error and adjusting ASWASH by DSWERR continues until the error changes signs;

$$\text{i.e., until} \quad \frac{\text{FLERR(IFL)}}{\text{FLERR(IFL-1)}} \quad \text{.LT. 0.0}$$

When this occurs, the next estimate of ASWASH is determined by linear interpolation between the last swash angle (ASWERR(IFL-1) and error (FLERR(IFL-1)) and the current swash angle (ASWERR(IFL)) and error (FLERR(IFL)) as shown in equation (3).

$$\text{ASWASH} = \text{ASWERR(IFL)} - (\text{ASWERR(IFL)-ASWERR(IFL-1)})* \frac{\text{FLERR(IFL)}}{(\text{FLERR(IFL)-FLERR(IFL-1)})} \tag{3}$$

The integer IFL is used to count the total number of swash angle iterations and the integer KINTRP is used to count the number of interpolations. The process is continued until:

(a) The flow error is reduced to less than 0.01 CIS (FLERR(IFL) .LT. 0.01); or,

(b) The incremental swash angle for the next ASWASH estimate is less than 0.001 degree; or,

(c) Two interpolations have taken place (KINTRP.EQ.2); or,

(d) Ten total iterations have occurred (IFL.GT.10).

Program execution continues with the existing error in all cases except (d) for which it is assumed that reasonable flow balance cannot be achieved and program execution stops.

4.3.1.3 Steady State Outlet Pressure Calculation - If QOVB is not greater than QMAXIY but the first estimate of swash angle is greater than maximum SWASH, steady state outlet pressure HPRESS is estimated by linear interpolation between QMAXY where HPRESS = PRESS and QMAXIY where HPRESS = 0.0 in accordance with equation (4).

$$\text{HPRESS} = \text{PRESS} - \text{PRESS} * \frac{(\text{QMAXY-QOVB})}{(\text{QMAXY-QMAXIY})} \tag{4}$$

HPRESS is adjusted to achieve steady state flow balance in the same manner described in paragraph 4.3.1.2 for the swash angle calculations. HPRESS is initially reduced (or increased) depending on the sign of the flow error until the zero error value is bracketed;

$$\text{i.e., until } \frac{\text{FLERR(IFL)}}{\text{FLERR(IFL-1)}} \text{ .LT. 0.0}$$

The next estimate of HPRESS is obtained by interpolating between the last pressure (HPRERR(IFL-1)) and error (FLERR(IFL-1))) and the current pressure (HPRERR(IFL)) and error (FLERR(IFL)) as shown in equation (5).

$$\text{HPRESS} = \text{HPRERR(IFL)} - (\text{HPRERR(IFL)-HPRERR(IFL-1)})* \frac{(\text{FLERR(IFL)})}{(\text{FLERR(IFL)-FLERR(IFL-1)})} \tag{5}$$

The criteria for steady state balance of outlet pressure is similar to that for the swash angle:

(a) (FLERR(IFL) .LT. 0.01); or,

(b) (KINTRP .EQ. 2); or

(c) (IFL .GT. 10).

In practice, rarely more than four iterations are required; the more usual number of iterations required to achieve steady state balance is two.

4.3.1.4 Flow Saturation - If the steady state flow demand exceeds the capability of the pump at a given RPM (QOVB.GT.QMAXIY), a warning message is printed (if write option 4 has been selected) and dynamic flow and pressure (Q(1) and P(1), respectively) are set to near-zero. Control passes back to the main program. If flow and/or pressure plotting options are selected, the resulting output plots show zero amplitude for dynamic flow and pressure.

4.3.2 Assumptions - Swash angle is assumed to vary linearly with flow demand between QZERO and QMAXY at a given RPM with steady state outlet pressure equal to the input data steady state circuit pressure. Similarly, steady state outlet pressure is assumed to vary linearly with flow demand between QMAXY and QMAXIY at a given RPM with swash angle equal to the input data maximum swash angle.

4.3.3 Computation Method - Linear interpolation is used to estimate swash angle or outlet pressure. Successive iterations reduce the resulting flow error to within specified limits.

4.3.4 Approximations - The equivalent steady state output flow at zero swash angle is approximated by straight line extrapolation of maximum flow at the starting and ending pump speeds to zero RPM.

4.3.5 Limitations - Near-zero dynamic flow and pressure values are returned to the main program if steady state overboard flow demand exceeds the pump capability at a given RPM. This exists as a calculated zero psi steady state outlet pressure condition. This should be interpreted as a meaningless situation, because the outlet lines would theoretically be less than full of fluid. The established relationships used for dynamic analysis do not apply.

4.3.6 Variable Names - See paragraph 4.1.6.

4.3.7 Steady State Swash Angle & Pressure Calculation - Listing

```
C
C     SECTION 2  SWASH ANGLE AND STEADY STATE OUTPUT PRESSURE CALCULATION
C
      PIA=DIAPIS**2*PI/4.
      CAVOL=0.0
      ORF=.65*SQRT(2./RHO)
      QWBEG=0.0
      QWEND=0.0
      QMAXI=0.0
  140 CONTINUE
      KINTRP=0
      IFL=1
      IF(Y.NE.WSTART)GOTO 146
      ASWASH=SWASH/57.3
      GOTO 175
  142 Y=WEND
      GOTO 179
  144 QZERO=QWEND-WEND*(QWEND-QWBEG)/(WEND-WSTART)
      ASWASH=SWASH/57.3
      HPRESS=0.0
      GOTO 179
  143 IF(IOPT(4).NE.4)GOTO 141
      WRITE(6,145)QZERO,QWBEG,WSTART,QWEND,WEND,QMAXI,WEND
  145 FORMAT(//,5X,50('*'),//,5X,'FLOW CONSTANTS USED FOR',
     1' SWASH ANGLE CALCULATION',//,10X,'QZERO= ',F10.6,' CIS AT    0.',
     2' RPM',/,10X,'QWBEG= ',F10.6,' CIS AT ',F5.0,' RPM',/ 10X,
     3'QWEND= ',F10.6 ' CIS AT ',F5.0,' RPM',/,10X,'QMAXI= ',
     4F10.6 ' CIS AT ',F5.0,' RPM AND ZERO OUTLET PRESSURE',
     5//,5X,50('*'),//)
  141 Y=WSTART
  146 QMAXY=QWEND-(WEND-Y)*(QWEND-QZERO)/WEND
      QMAXIY=Y*QMAXI/WEND
      IF(QOVB.GT.QMAXIY)GOTO 177
      ASWASH=(SWASH-SWASH*(QMAXY-QOVB)/(QMAXY-QZERO))/57.3
      IF(ASWASH.LT.SWASH/57.3)GOTO 178
      ASWASH=SWASH/57.3
      IF(QOVB.GT.QMAXIY)GOTO 177
  147 IF(QOVB.EQ.0.0)GOTO 175
      HPRESS=PRESS-PRESS*(QMAXY-QOVB)/(QMAXY-QMAXIY)
      GOTO 179
  177 IF(IOPT(4).NE.4)GOTO 335
      WRITE(6,257)Y
      WRITE(6,263)
      GOTO 335
  175 HPRESS=PRESS
```

4.4 SECTION 3 - PISTON PRECOMPRESSION CALCULATION

Section 3 calculates the cylinder pressure which exists just before the cylinder slot starts to open to the valve plate pressure slot. This pressure results from compressing the fluid in the cylinder during that portion of block rotation when the cylinder slot is closed by the valve plate between the suction and pressure slots.

4.4.1 Math Model - Piston motion results from rotating the cylinder block with the piston shoes held against the angled swash plate. Each swash plate angle component (the fixed cross angle and the controlled swash angle) contributes to total piston motion, which may be defined as the sum of two sinusoidal motions as shown in Figure 4-7. Piston positions at the start and end of incremental stroke DX and their respective pressures are defined as shown in Figure 4-8.

4.4.2 Assumptions - The pressure dependent factor for leakage out of the cylinder (with cylinder pressure greater than case pressure) is as shown in equation (1).

$$\text{PLEAK} = \frac{\text{TLEAK}}{\text{NAPP*PRESS}} \tag{1}$$

The pressure dependent factor for leakage into the cylinder is assumed to be as shown in equation (2).

$$\text{SLEAK} = -\frac{\text{TLEAK}}{\text{NAPS*SQRT(CSPRES)}} \tag{2}$$

The piston is assumed to be completely filled with fluid at the start of calculation, with piston pressure PISPR( ) equal to the input data steady state inlet pressure.

4.4.3 Computation Method - The calculation starts with the cylinder centerline angle THETA at the last index position before the suction slot closes, NDEG = NSUCL. Precompression pressure is calculated in 1/2 degree increments of cylinder block rotation. Calculated values are stored in the array PISPR(NDEG).

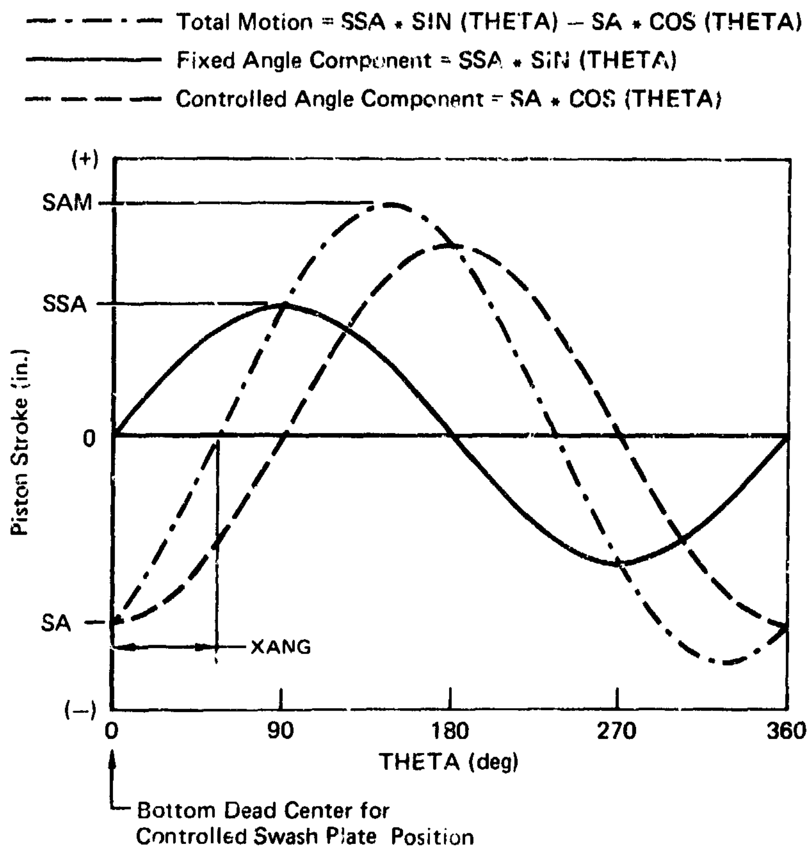


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FIGURE 4-7
PISTON MOTION

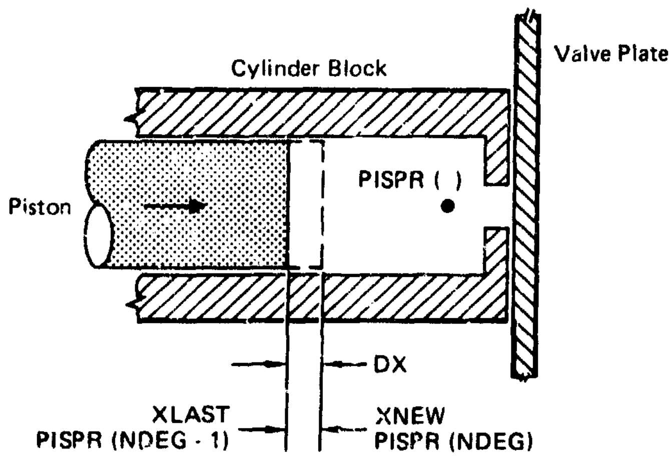

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FIGURE 4-8
PISTON PRECOMPRESSION

Bulk modulus is calculated at the beginning of each calculation step. The pressure rise due to compression is calculated and is added to the final piston pressure from the preceding step. Net leakage volume DLEAK is calculated using the applicable pressure dependent leakage factor PLEAK or SLEAK. Bulk modulus is recalculated for the new piston pressure (after compression) and the differential pressure increment resulting from a volumetric change equivalent to DLEAK is calculated. Net piston pressure at the end of the calculation step is given by equation (3).

$$\text{PISPR(NDEG)} = \text{PISPR(NDEG-1)} + \Delta P \text{ comp} - \Delta P \text{ leak} \tag{3}$$

The process is repeated until the last index position before the pressure slot starts to open is reached, NDEG = NPRSOP.

4.4.4 Approximations - Net leakage volume out of (or into) the cylinder is approximated by the pressure dependent leakage factors and is assumed to take place at the end of the incremental compression stroke.

4.4.5 Limitations - Dynamic effects on the initial inlet pressure are not included.

4.4.6 Variable Names - See paragraph 4.1.6.

4.4.7 Piston Precompression Calculation - Listing

```
C
C     SECTION 3 - PISTON PRECOMPRESSION CALCULATION
C
  179 CONTINUE
      SA=RBORC*TAN(ASWASH)
      SSA=RBORC*TAN(ANGCR/57.3)
      SAN=SQRT(SA**2+SSA**2)
      DT=ALNC/(Y*6.)
      PLEAK=TLEAK/(PRESS*NAPP)
      SLEAK=-(TLEAK/NAPS/SQRT(CSPRES))
      NDEG=NSUCL
      PISPR(NDEG)=LPRESS
      THETA=(NDEG/2.-0.5)/57.3
      XLAST=SSA*SIN(THETA)-SA*COS(THETA)
  162 NDEG=NDEG+1
      IF(NDEG.EQ.721)NDEG=1
      THETA=(NDEG/2.-0.5)/57.3
      XNEW=SSA*SIN(THETA)-SA*COS(THETA)
      DVOL=(XNEW-XLAST)*PIA
      VA=POVOL-XLAST*PIA
      NSTART=NDEG-1
      IF(NSTART.EQ.0)NSTART=720
      BULKP=BULK+12.*(PISPR(NSTART)-PRESS)
      PISPR(NDEG)=PISPR(NSTART)+BULKP*DVOL/VA
      DLEAK=(PISPR(NDEG)-CPRESS)*PLEAK*DT
      IF(PISPR(NDEG).LT.CPRESS)DLEAK=SQRT(CPRESS-PISPR(NDEG))*
     1SLEAK*DT
      VALEAK=POVOL-XNEW*PIA
      BULKP=BULK+12.*(PISPR(NDEG)-PRESS)
      DPLEAK=BULKP*DLEAK/VALEAK
      PISPR(NDEG)=PISPR(NDEG)-DPLEAK
      IF(PISPR(NDEG).GT.0.01)GOTO 164
      CAVOL=CAVOL-DVOL+DLEAK
      PISPR(NDEG)=0.01
      GOTO 163
  164 CAVOL=0.0
  163 XLAST=XNEW
      IF(IOPT(2).NE.2)GOTO 167
      IF(Y.NE.YOPT(1))GOTO 157
      IF(NDEG.EQ.NSUCL+1)WRITE(6,160)
  160 FORMAT('1',10X,'**** PRECOMPRESSION PRESSURES ****',//,
     +5X,'NDEG   THETA     PISTON',8X,'BULK',7X,'CAVOL      PISPR(NDEG)',/,
     +19X,'POSITION   MODULUS',/)
      WRITE(6,161)NDEG,THETA*57.3,XNEW,BULKP,CAVOL,PISPR(NDEG)
  161 FORMAT(5X,I4,2X,F5.1,2X,F10.6,2X,F10.2,2X,F10.6,2X,F10.4)
  157 IF(NDEG.EQ.NPRSOP)GOTO 166
      GOTO 162
  166 CONTINUE
      CAVOLD=CAVOL
      DO 150 N=1,81
      PPT(N)=0.0
  150 CONTINUE
      KKN=1
      KK=1
  170 CONTINUE
```

4.5 SECTION 4 - PUMP OUTPUT FLOW CALCULATION

Section 4 calculates the total output flow for one pumping cycle, i.e., 40 degrees of cylinder block rotation for a nine piston pump. The primary output flow results from displacement of the piston in the cylinder. Primary flow is modified by such factors as the pressure rise due to compression, the throttling effect of orifice flow through the valve opening, and flow losses due to leakage. Other factors such as cavitation and the influence of dynamic pressure on the steady state outlet pressure are also considered.

4.5.1 Math Model

4.5.1.1 Primary Flow - Piston position is determined by the angular position of cylinder block rotation THETA and by the swash plate angle components. From Section 3, piston position X is defined by equation (1);

$$X = SSA*SIN(THETA)-SA*COS(THETA) \quad (1)$$

which may also be expressed as:

$$X = SAM*SIN(THETA-XANG) \quad (2)$$

where $SAM = SQRT(SSA**2 + SA**2)$ (3)

$$XANG = TAN^{-1}(SA/SSA) \quad (4)$$

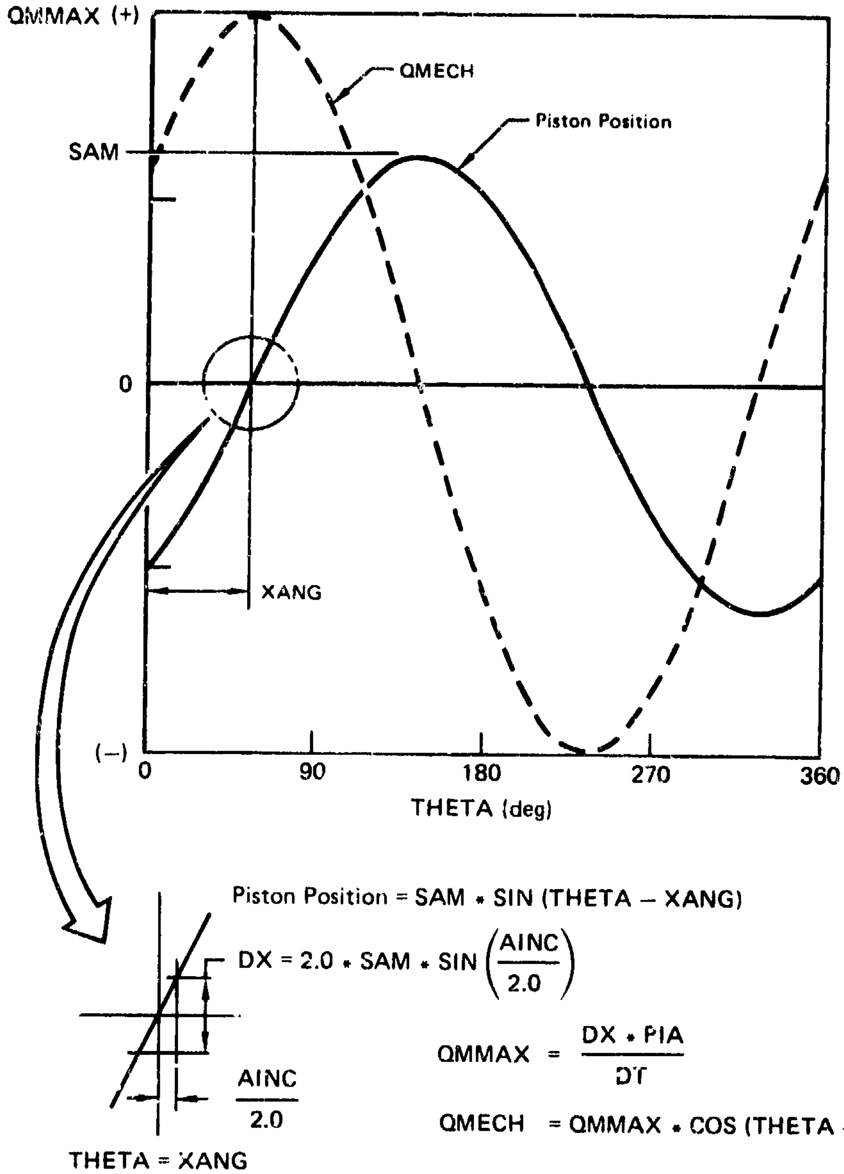
The primary flow resulting from displacement of the piston is defined as QMECH and lags piston position by 90 degrees as shown in Figure 4-9. Peak QMECH occurs when THETA=XANG and is determined as follows.

Select a small increment of cylinder block rotation, AINC

Let THETA = XANG+AINC/2.0

So that piston displacement DX over the increment AINC is given by:

$$DX = 2.0*SAM*SIN(AINC/2.0). \quad (5)$$



Piston Position = SAM * SIN (THETA – XANG)

$$DX = 2.0 * SAM * SIN\left(\frac{AINC}{2.0}\right)$$

$$QMMAX = \frac{DX * PIA}{DT}$$

QMECH = QMMAX * COS (THETA – XANG)

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FIGURE 4-9
OMECH CALCULATION

The volumetric displacement is DX*PIA

So that average flow over the time DT equals DX*PIA/DT.

DX is selected so that the average flow just calculated occurs at THETA = XANG so that the peak value of QMECH is:

$$QMMAX = DX*PIA/DT \tag{6}$$

Therefore, for any angular position of the cylinder block:

$$QMECH = QMMAX*COS(THETA-XANG) \tag{7}$$

4.5.1.2 Output Flow - Figure 4-10 illustrates the parameters used to calculate output flow, QOUT. Equations (8) and (9) define QOUT.

$$QOUT = QMECH - \Delta Qleak - \Delta Qcomp \tag{8}$$

$$QOUT = CD*VAREA(NDEG)*SQRT(2.*(PISPR(NDEG)- (HPRESS+PPT(NKM))/RHO) \tag{9}$$

Where: QMECH is the primary flow from paragraph 4.5.1.1

ΔQleak is the flow loss due to leakage

ΔQcomp is the flow loss due to fluid compression

CD is the orifice discharge coefficient

VAREA(NDEG) is the valve area from Section 1

PISPR(NDEG) is the piston chamber pressure at the end of incremental stroke DX

HPRESS+PPT(NKM) is the net outlet pressure

RHO is the fluid density

Consider equation (8). The leakage from the cylinder to case is estimated from the differential pressure at the end of the incremental stroke DX and the leakage factor LEAK, so that:

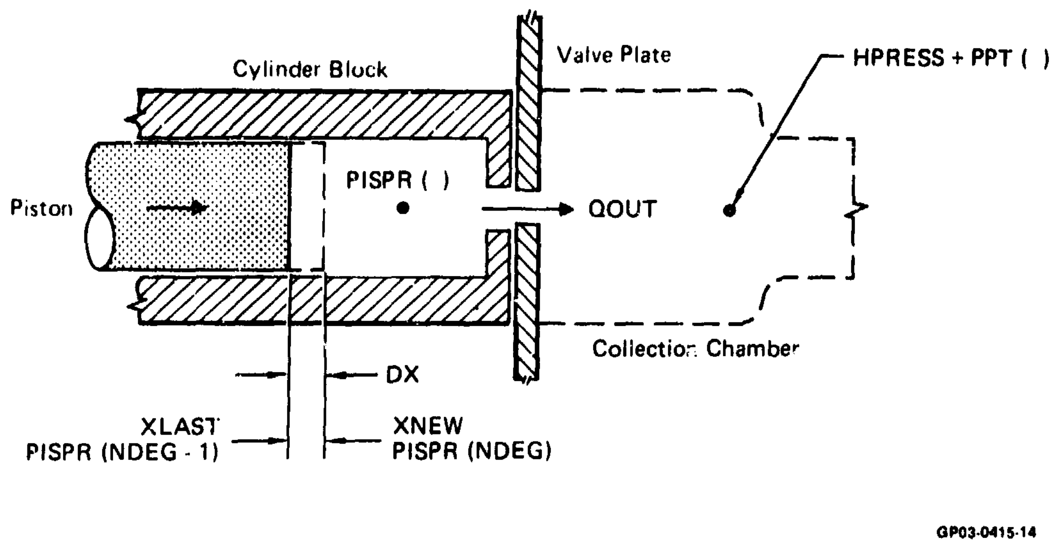


FIGURE 4-10
QOUT CALCULATION

$$\Delta Qleak = (PISPR(NDEG)-CPRESS)*LEAK \tag{10}$$

The equivalent flow loss due to compressibility is derived from the bulk modulus formula:

$$\beta = \frac{\Delta P*Volume}{\Delta Volume} \tag{11}$$

Rearranging and considering ΔVolume over the time increment ΔT yields:

$$\Delta Qcomp = \frac{\Delta Volume}{\Delta T} = \frac{\Delta P*Volume}{\beta*\Delta T} \tag{12}$$

Since ΔP over the incremental stroke DX is the difference between PISPR(NDEG) and PISPR(NDEG-1),

$$\Delta Qcomp = (PISPR(NDEG)-PISPR(NDEG-1))*\frac{VA}{BULKP*DT} \tag{13}$$

Where VA is the original cylinder volume at the start of the incremental stroke DX. Substituting equations (10) and (13) in equation (8) yields:

$$\begin{aligned} QOUT=QMECH-&(PISPR(NDEG)-CPRESS)*LEAK - \\ &(PISPR(NDEG)-PISPR(NDEG-1))*\frac{VA}{BULKP*DT} \end{aligned} \tag{14}$$

Now consider equation (9). Solving for PISPR(NDEG) yields:

$$PISPR(NDEG) = \frac{RHO*QOUT**2}{2*(CD*VAREA(NDEG))**2} + (HPRESS+PPT(NKM)) \tag{15}$$

Let $$LK1 = \frac{1}{LEAK + \frac{VA}{BULKP*DT}} \tag{16}$$

$$LK2 = QMECH + CPRESS*LEAK+PISPR(NDEG-1)* \frac{VA}{BULKP*DT} \tag{17}$$

$$\frac{1}{LK3} = \frac{RHO}{2*(CD*VAREA(NDEG))**2} \tag{18}$$

Substituting equations (16) and (17) in (14) and rearranging yields:

$$QOUT = LK2 - \frac{PISPR(NDEG)}{LK1} \tag{19}$$

Similarly, substituting equation (18) in (15) yields:

$$PISPR(NDEG) = \frac{QOUT^2}{LK3} + (HPRESS + PPT(NKM)) \tag{20}$$

For each calculation step the last calculated value of QOUT is used for the initial estimate of flow, QEST. Equation (20) is used to compute an initial trial value of piston pressure, PTRIAL(1). This value is substituted for PISPR(NDEG) in equation (19) to produce QTRIAL (1).

The QTRIAL( ) from equation (19) is compared to the QEST used in equation (20). If the difference is significant, QITERR.GE.0.5%, QEST is updated by averaging the last and current QTRIAL( ) values. PTRIAL( ) and QTRIAL( ) values are recalculated using the appropriate substitutions in equations (19) and (20). The iteration process is repeated until flow balance between equations (19) and (20) is achieved. QOUT and PISPR(NDEG) are set equal to the balanced values of QTRIAL( ) and PTRIAL( ), respectively.

4.5.2 Assumptions

(a) Flow through the cylinder/valve plate slots is described by the orifice equation. The discharge coefficient used is developed for a circular orifice with an area equivalent to VAREA(NDEG).

(b) Leakage is assumed to take place at the end of the incremental stroke, DX.

(c) The flow resulting from piston displacement (QMECH) is assumed to lag piston position by 90 degrees, regardless of outlet pressure dynamic effects.

(d) The flow contribution of all cylinders that are exposed to the pressure slot may be summed to produce total time dependent output flow. It is assumed that there is no interactive effect between cylinders.

4.5.3 Computation Method - Flow is calculated for one piston as its cylinder traverses the valve plate pressure slot in 1/2 degree increments of cylinder block rotation.

Figure 4-11 illustrates the flow calculation procedure. When the cylinder slot of Piston 1 just begins to open to the valve plate pressure slot, it can be seen that the next piston (Piston 2) has just passed the fully open position. Piston 3 is nearing maximum stroke and Piston 4 is starting to retract into its cylinder. For the example shown, the next cylinder has just closed to the pressure slot so that it and the remaining pistons have no active bearing on the output flow calculation.

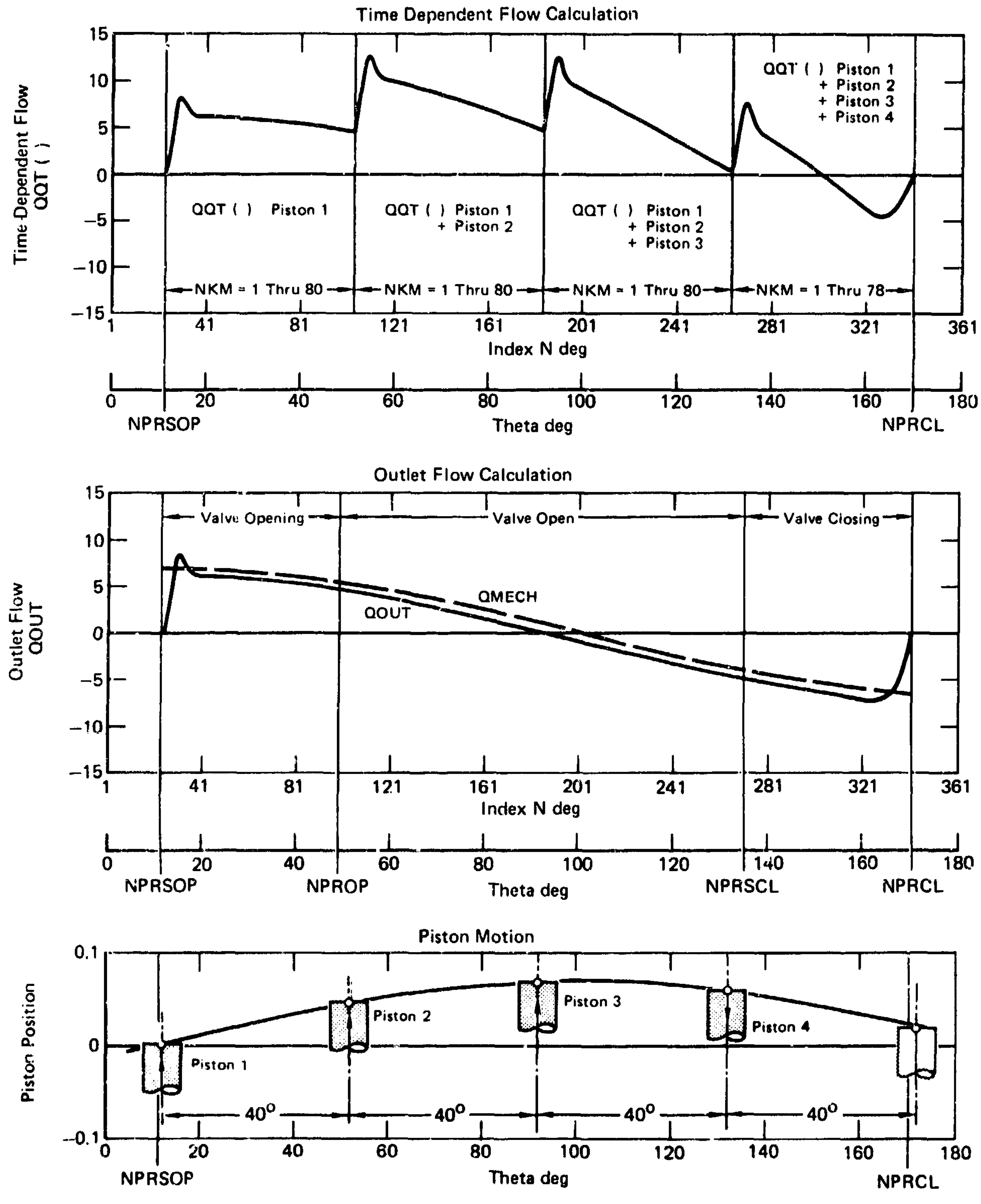


FIGURE 4-11
FLOW CALCULATION

The QMECH and QOUT calculations described in paragraph 4.5.1 are performed for Piston 1 starting with the cylinder centerline at the first index position after valve opening begins, NDEG = NPRSOP+1. The computation proceeds in 1/2 degree increments of cylinder block rotation. The calculated value of QOUT for each index position is stored in the time dependent flow array, QQT(NKM).

The time dependent counter NKM is reset after each 40 degrees of cylinder block rotation (80 index positions) so that the array QQT(NKM) is filled first with the flow contribution of Piston 1. The flow contribution of Piston 2 is summed with that for Piston 1, time step for time step. That total is summed with the flow contributions of Pistons 3 and 4 in the same manner. The calculation proceeds until the index counter NDEG = NPRCL. At the completion of the process, the flow contribution of all active pumping pistons is accounted for. The final values in the array QQT(NKM) represent the total pump output flow waveform for one pumping cycle.

4.5.4 Approximations - Peak QMECH is approximated for the time step DT by using the average flow over the angular increment of cylinder block rotation, AINC, at the position when the slope of piston travel is maximum. The angular increment AINC is sufficiently small so that errors resulting from this approximation are considered to be negligible.

4.5.5 Limitations - None.

4.5.6 Variable Names - See paragraph 4.1.6.

4.5.7 Section 4 - Pump Output Flow Calculation - Listing

```
C
C     SECTION 4 - PUMP OUTPUT FLOW CALCULATION
C
      DX=2.*SAM*SIN(AINC/(2.*57.3))
      QMMAX=DX*PIA/DT
      XANG=ATAN2(SA,SSA)
      CAVOL=CAVOLD
      DO 180 NKM=1,81
      QQT(NKM)=0.0
  180 CONTINUE
      NDEG=NPRSOP
      THETA=(NDEG/2.-0.5)/57.3
      XLAST=SSA*SIN(THETA)-SA*COS(THETA)
      NKM=0
      QOUT=0.0
      QEST=QOUT
      CD=0.65
      NBEG=NPRSOP+1
      DO 190 NDEG=NBEG,NPRCL
      NKM=NKM+1
      IF(NKM.EQ.81)NKM=1
      THETA=(NDEG/2.-0.5)/57.3
      XNEW=SSA*SIN(THETA)-SA*COS(THETA)
      DX=XNEW-XLAST
      VA=POVOL-XLAST*PIA
      XLAST=XNEW
      QMECH=QMMAX*COS(THETA-XANG)
      IF(CAVOL.GT.0.0)GOTO 187
      BULKP=BULK+12.*(PISPR(NDEG-1)-PRESS)
      LEAK=SLEAK
      IF(PISPR(NDEG-1).GE.CPRESS)LEAK=PLEAK
      LK1=1./((VA/(BULKP*DT))+LEAK)
      LK2=QMECH+VA*PISPR(NDEG-1)/(BULKP*DT)+CPRESS*LEAK
      LK3=2.*(CD*VAREA(NDEG))**2/RHO
      QEQUAT=1.
      ITER=1
  181 CONTINUE
      PTRIAL(ITER)=QEST*ABS(QEST)/LK3+
     1(HPRESS+PPT(NKM))
      QTRIAL(ITER)=LK2-PTRIAL(ITER)/LK1
      IF(ITER.LE.2)GOTO 186
      IF(ITER.GT.3)GOTO 185
      IF(QTRIAL(ITER).LE.QTRIAL(ITER-1)
     +.AND.QTRIAL(ITER).GE.QTRIAL(ITER-2)
     +.OR.QTRIAL(ITER).LE.QTRIAL(ITER-2)
     +.AND.QTRIAL(ITER).GE.QTRIAL(ITER-1))GOTO 185
      QEQUAT=2.
      ITER=1
      QEST=QOUT
  182 CONTINUE
```

4.5.7 (Continued)

```
      PTRIAL(ITER)=LK1*(LK2-QEST)
      QTRIAL(ITER)=SQRT(LK3*ABS(PTRIAL(ITER)-(HPRESS+PPT(NKM))))
      IF(PTRIAL(ITER).LT.(HPRESS+PPT(NKM)))QTRIAL(ITER)=-QTRIAL(ITER)
      IF(ITER.EQ.1)GOTO 186
  185 CONTINUE
      QITERR=ABS(QTRIAL(ITER)-QTRIAL(ITER-1))
      IF(QTRIAL(ITER-1).NE.0.0)QITERR=ABS(QITERR/QTRIAL(ITER-1))
      IF(QITERR.LT.0.005)GOTO 188
  186 CONTINUE
      QEST=(QTRIAL(ITER)+QEST)/2.
      IF(ITER.LT.3)QEST=QTRIAL(ITER)
      IF(ITER.EQ.10)GOTO 188
      ITER=ITER+1
      IF(QEQUAT.EQ.2.)GOTO 182
      GOTO 181
  187 CONTINUE
      LEAK=SLEAK
      QOUT=-(CD*VAREA(NDEG)*
     1SQRT(2.*(HPRESS+PPT(NKM))/RHO))
      PISPR(NDEG)=0.01
      CAVOL=CAVOL-DX*PIA+SLEAK*DT*CPRESS+QOUT*DT
      IF(CAVOL.LE.0.0)CAVOL=0.0
  188 CONTINUE
      QOUT=QTRIAL(ITER)
      PISPR(NDEG)=PTRIAL(ITER)
      QEST=QOUT
      QQT(NKM)=QOUT+QQT(NKM)
      IF(IOPT(3).NE.3)GOTO 190
      IF(Y.NE.YOPT(2))GOTO 190
      IF(NDEG.EQ.NBEG)WRITE(6,183)
  183 FORMAT('1',10X,'**** OUTLET FLOW VALUES ****',//,
     +5X,'KK NKM NDEG   THETA     PISTON',6X,'PISTON',7X,'QOUT',
     +6X,'QQT(NKM)',6X,'CAVOL',5X,'PISPR(NDEG)',/,
     +26X,'POSITION',6X,'FLOW',/)
      WRITE(6,184)KK,NKM,NDEG,THETA*57.3,XNEW,QMECH,QOUT,QQT(NKM),
     +CAVOL,PISPR(NDEG),PPT(NKM)
  184 FORMAT(5X,I2,1X,I3,1X,I4,2X,F5.1,2X,F10.6,2X,3(F10.4,2X),F10.6,2X,
     +F10.4,2X,F10.4)
  190 CONTINUE
      QQT(81)=QQT(1)
```

4.6 SECTION 5 - FOURIER ANALYSIS OF PUMP OUTLET FLOW

Section 5 performs a mathematical harmonic analysis of the time dependent output flow waveform, QQT( ), from Section 4. Dynamic flow is calculated over the pumping cycle period for each harmonic from the fundamental through the input data harmonic of interest.

Section 5 also contains the calculations for establishing the flow constants and steady state swash angle and output pressure balance iterations used in Section 2. The steady state balance calculations are bypassed during dynamic balancing in Section 6.

4.6.1 Math Model - The Fourier Analysis calculations used in this section are from a modified IBM subroutine (Reference 2). COSINE and SINE amplitudes from Fourier Analysis are combined to form complex dynamic output flow, FQ1(-,-). Dynamic flow is phase related to the pumping cycle established by the time dependent outlet flow waveform.

Figure 4-12(a) compares the dynamic output flow that results from Fourier Analysis to the calculated time dependent output flow from Section 4. Figure 4-12(b) compares the first calculation of dynamic flow, FQ1(1,1), to the last value, FQ1(last), which results from dynamic pressure-flow balance in Section 6. The dynamic pressure that produces this change in flow is also shown.

4.6.2 Assumptions - None

4.6.3 Computation Method - Fourier Analysis

4.6.4 Approximations - None

4.6.5 Limitations - None

4.6.6 Variable Names - See paragraph 4.1.6.

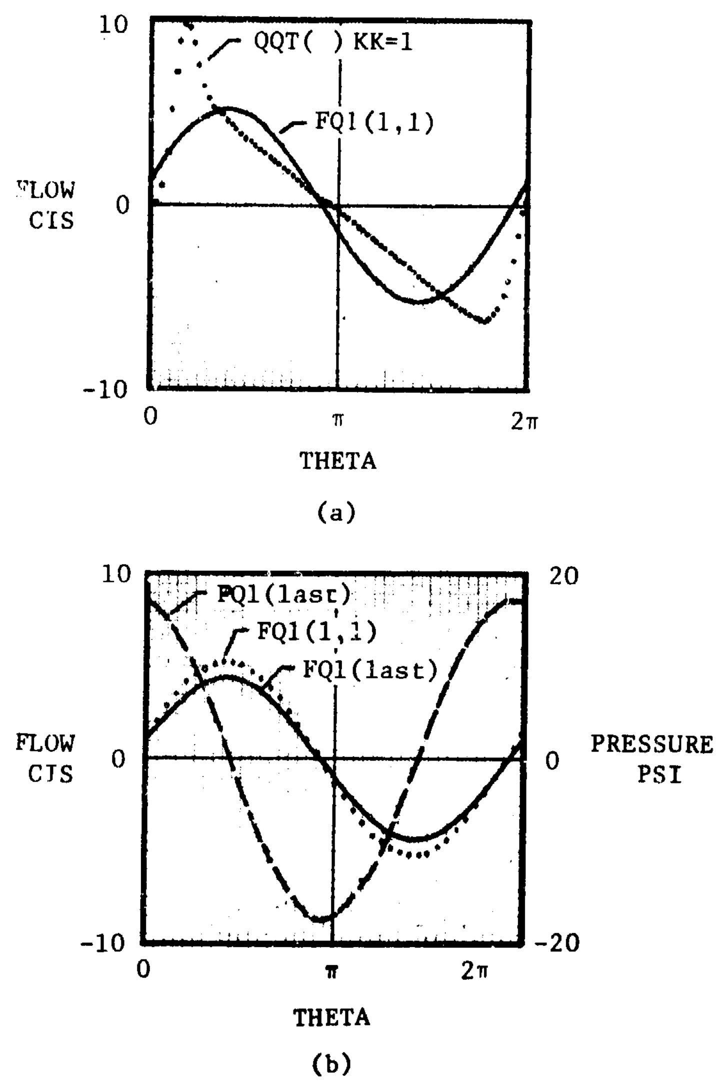
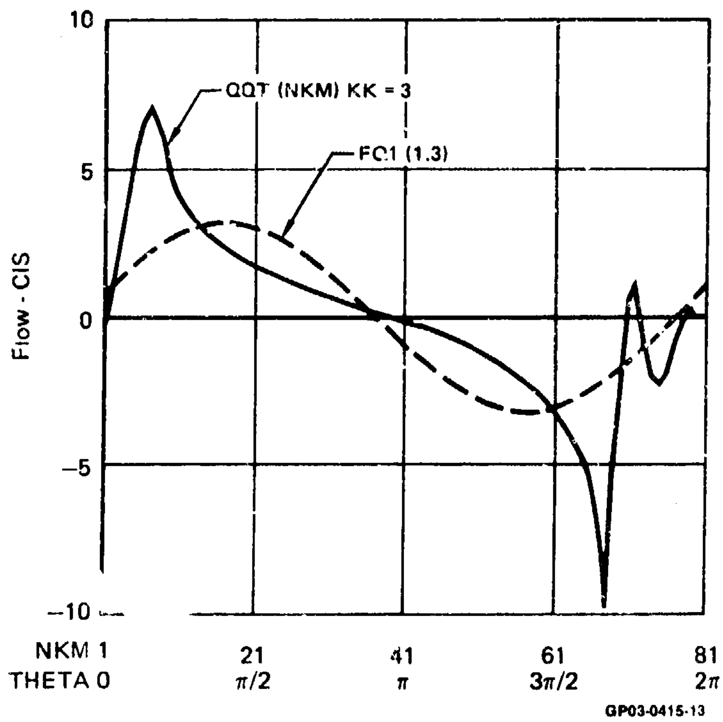


FIGURE 4-12
DYNAMIC FLOW



10
5
0
−5
−10
QQT (NKM) KK = 3
FC1 (1,3)
Flow - CIS
NKM 1
21
41
61
81
THETA 0
π/2
π
3π/2
2π
GP03-0415-13

4.6.7 Section 5 - Fourier Analysis of Pump Output Flow - Listing

```
C      SECTION 5- FOURIER ANALYSIS OF PUMP OUTPUT FLOW
C
  201 AN=360./PISTNO
      N=AN+.0001
      COEF=2.0/(2.0*AN+1.0)
      C1=PI*COEF
      S1 =SIN(C1)
      C1 =COS(C1)
      S =0.0
      C =1.0
      FNTZ=QQT(1)
      J =1
  210 U2=0.0
      U1=0.0
      I=2*N+1
C       FORM FOURIER COEFFICIENTS RECURSIVELY
  220 U0=QQT(I)+2.0*C*U1-U2
      U2=U1
      U1=U0
      I=I-1
      IF(I-1) 230,230,220
  230 QQFC(J)=COEF*(FNTZ+C*U1-U2)
      QQFS(J)=COEF*S*U1
      IF(J-NHARM-1) 240,250,250
  240 U3=C1*C-S1*S
      S =C1*S+S1*C
      C =U3
      J = J+1
      GO TO 210
  250 QQFC(1)=QQFC(1)*0.5
C
C      ESTABLISH QWBEG AND QWEND
C
      IF(KK.NE.1)GOTO 255
      IF(QWBEG.NE.0.0.AND.QWEND.NE.0.0.AND.QMAXI.NE.0.0)GOTO 252
      IF(QWBEG.NE.0.0)GOTO 251
      QWBEG=QQFC(1)
      IF(QWBEG.EQ.0.0)QWBEG=0.0001
      GOTO 142
  251 IF(QWEND.NE.0.0)GOTO 261
      QWEND=QQFC(1)
      GOTO 144
  261 QMAXI=QQFC(1)
      GOTO 148
  252 CONTINUE
      FLERR(IFL)=QQFC(1)-QOVB
      ASWERR(IFL)=ASWASH
      HPRERR(IFL)=HPRESS
      IF(IOPT(4).NE.4)GOTO 249
      IF(IFL.EQ.1)WRITE(6,267)Y
```

4.6.7 <u>(Continued)</u>

```
  267 FORMAT('1',90('*'),/,33X,'DATA FOR',F6.0,' RPM',///,10X,
     1'*** SWASH ANGLE-STEADY STATE OUTLET PRESSURE CALCULATION ***',
     2//,15X,'IFL',5X,'SWASH',6X,'ST.STATE',6X,'FLOW',6X,'ST.STATE',/,
     323X,'ANGLE',8X,'FLOW',7X,'ERROR',6X,'PRESSURE',//)
      WRITE(6,253)IFL,ASWASH*57.3,QQFC(1),FLERR(IFL),HPRERR(IFL)
  253 FORMAT(15X,I3,2X,4(F10.4,2X))
  249 IF(ABS(FLERR(IFL)).LT.0.01)GOTO 255
      IF(ASWASH.LT.SWASH/57.3)GOTO 256
  263 FORMAT(/,15X,'*** CAUTION ***-FLOW DEMAND EXCEEDS PUMP',
     1' CAPABILITY AT THIS RPM',/)
  248 CONTINUE
      IF(IFL.EQ.1.AND.KINTRP.LT.1)GOTO 266
      IF(FLERR(IFL)/FLERR(IFL-1).GT.0.0)GOTO 266
  262 HPRESS=HPRERR(IFL)-(HPRERR(IFL)-HPRERR(IFL-1))*
     1FLERR(IFL)/(FLERR(IFL)-FLERR(IFL-1))
      IF(HPRESS.LT.0.0)HPRESS=0.0
      KINTRP=KINTRP+1
      GOTO 254
  266 IF(QQFC(1).LT.QOV3)GOTO 268
      HPRESS=PRESS
      GOTO 254
  268 HPRESS=.50*HPRESS
      GOTO 254
  256 CONTINUE
      IF(KINTRP-1)259,258,255
  259 IF(IFL.EQ.1)GOTO 257
      IF(FLERR(IFL)/FLERR(IFL-1).GT.0.0)GOTO 257
  258 ASWASH=ASWERR(IFL)-(ASWERR(IFL)-ASWERR(IFL-1))*
     1FLERR(IFL)/(FLERR(IFL)-FLERR(IFL-1))
      KINTRP=KINTRP+1
      GOTO 254
  257 DSWERR=(SWASH-SWASH*(QMAXY-(QZERO+ABS(FLERR(IFL))))
     1/(QMAXY-QZERO))/57.3
      IF(ABS(DSWERR)*57.3.LT.0.001)GOTO 255
      ASWASH=ASWASH-DSWERR*FLERR(IFL)/ABS(FLERR(IFL))
  254 IFL=IFL+1
      IF(IFL.GT.10) STOP 2002
      IF(ASWASH.GE.SWASH/57.3)GOTO 179
      GOTO 178
C
C       COMPUTE HARMONIC FLOWS FROM FOURIER ANALYSIS
C
  255 DO 260 I=1,NHARM
      FQ1(I,KK) = CMPLX(QQFC(I+1), QQFS(I+1))
  260 CONTINUE
?
```

4.7 SECTION 6 - OUTLET PRESSURE - FLOW BALANCE CALCULATION

Section 6 estimates pump internal impedance and the complex outlet pressure based on the system load impedance and complex dynamic pump flow.

4.7.1 Math Model - The pump is modeled as an alternating flow source in parallel with a shunt impedance. The load seen by the flow source consists of the system load impedance for the harmonic of interest, ZIP(I), and its own parallel shunt impedance, ZO.

The first calculation of dynamic flow assumes that dynamic pressure equals zero. This flow value, FQ1(I,1), is used with the circuit impedance, ZIP(I), to produce a test value for dynamic pressure, PQ1(I,1), as shown in equation (1).

$$PQ1(I,1) = FQ1(I,1)*ZIP(I) \tag{1}$$

Time dependent pressure values are calculated in Section 7 and are then used in Section 4 to calculate time dependent flow for KK = 2. The value of dynamic flow that results from Fourier Analysis is used to define shunt impedance and flow, ZO and QSHUNT(1), respectively, in accordance with equations (2) and (3).

$$ZO = \frac{PQ1(I,1)}{FQ1(I,2)-FQ1(I,1)} \tag{2}$$

$$QSHUNT(1) = \frac{PQ1(I,1)}{ZO} \tag{3}$$

A second test value for dynamic pressure equal to one-half PQ1(I,1) is used to produce a third dynamic flow value, FQ1(I,3). Shunt impedance and flow values (RE and QSHUNT(2), respectively) are calculated for this set of dynamic flow and pressure conditions.

A Norton equivalent electrical circuit analogy is used to calculate dynamic pressure for the pump/circuit model. For a given circuit consisting of a current source and impedance, an equivalent circuit may be constructed such that the potential, E_o, across the impedances may be determined by:

$$E_o = \Delta I * \frac{R_o * R_e}{R_o + R_e} \tag{4}$$

where ΔI is the current difference
R_o is the impedance of the original circuit
R_e is the impedance of the equivalent circuit

Applying the analogy to the computed shunt impedance and flow values for the pump results in the following definition of dynamic pressure:

$$PQ1(I,3) = (QSHUNT(1) + QSHUNT(2))* \frac{ZO*RE}{ZO+RE} \tag{5}$$

The entire process is repeated for each harmonic through the harmonic of interest.

4.7.2 Assumptions - The pump is assumed to behave as an acoustic flow source with a parallel shunt impedance.

4.7.3 Computation Method - Electrical network analogy based on a Norton equivalent circuit is used to calculate pump internal impedance.

4.7.4 Approximations - None

4.7.5 Limitations - None.

4.7.6 Variable Names - See paragraph 4.1.6.

4.7.7 <u>Section 6 - Pressure Flow Balance - Listing</u>

```
C
C     SECTION 6 - OUTLET PRESSURE-FLOW BALANCE CALCULATION
C
  270 CONTINUE
      I=KKN
      IF(KK-2)280,290,300
  280 CONTINUE
      PQ1(I,1)=FQ1(I,1)*ZIP(I)
      P3=PQ1(I,1)
      GOTO 320
  290 CONTINUE
      ZO=PQ1(I,1)/(FQ1(I,2)-FQ1(I,1))
      ZE=ZO*ZIP(I)/(ZO+ZIP(I))
      QSHUNT(1)=PQ1(I,1)/ZO
      PQ1(I,2)=PQ1(I,1)*0.5
      P3=PQ1(I,2)
      GOTO 320
  300 CONTINUE
      J=KKN
      RE=PQ1(J,2)/(FQ1(J,3)-FQ1(J,1))
      ZE=RE*ZIP(J)/(RE+ZIP(J))
      QSHUNT(2)=PQ1(J,2)/RE
      PQ1(J,3)=(QSHUNT(1)+QSHUNT(2))*ZO*RE/(ZO+RE)
      IF(IOPT(4).NE.4)GOTO 310
      WRITE(6,322)KK,RE,ZIP(J),ZE
  310 CONTINUE
      KKN=KKN+1
      IF(KKN.GT.NHARM)GOTO 335
      KK=1
      FQ1(KKN,1)=FQ1(KKN,3)
      GOTO 270
  320 CONTINUE
      IF(IOPT(4).NE.4)GOTO 325
      IF(KK.EQ.1)WRITE(6,269)ASWASH*57.3,QOVB,PRESS,QMAXY,QMAXIY
  269 FORMAT(//,10X,'**** FLOW CALCULATION DATA ****',//,
     +10X,'SWASH ANGLE',5X,'=',F10.4,' DEG',/,
     +10X,'FLOW DEMAND',5X,'=',F10.4,' CIS',/,
     +10X,'FLOW CAPABILITY ',/,
     +13X,'AT',F6.0,' PSI =',F10.4,' CIS',/,
     +14X,'AT ZERO PSI =',F10.4,' CIS')
```

4.8 SECTION 7 - RECONSTRUCT TIME DEPENDENT OUTLET PRESSURE

Section 7 computes the time dependent output pressure from each estimate of complex dynamic output pressure (Section 6). Complex pump output balanced flow and pressure are stored for returning to the main program. If inlet analysis is not required, control is returned to the main program.

4.8.1 Math Model - The array PPT( ) is used to store 80 values of time dependent pressure for the pump cycle period. The array values are corrected with each test pressure and harmonic so that PPT( ) contains the total of the balanced pressures for each harmonic through the selected harmonic.

Each PPT( ) value consists of the amplitudes of the last complex dynamic pressure, P3, by the relationship:

$$\text{PPT( )} = \text{REAL(P3)} * \text{COS(THETA)} + \text{AIMAG(P3)} * \text{SIN (THETA)} \quad (1)$$

Where THETA is the time dependent cycle position for each increment of a 2Π cycle period in multiples of the harmonic number.

PPT( ) is initialized to zero for each RPM.

The inlet code number NINLT is tested when dynamic balancing is complete; i.e., KK = 3 and I = the harmonic number of interest. If inlet analysis is not required (NINLT. EQ. (21), control passes to the end of Section 13 where the values of P(1) and Q(1) are returned to the main program. If (NINLT. NE. 121) program execution continues with Section 8.

Representative values of the P(1) and Q(1) waveforms are shown in Figure 4-13.

4.8.2 Assumptions - Time dependent outlet pressure may be described as the sum of the COSINE and SINE amplitude of complex dynamic pressure determined in Section 6.

4.8.3 Computation Method - Not applicable.

4.8.4 Approximations - None.

4.8.5 Limitations - None.

4.8.6 Variable Names - See paragraph 4.1.6.

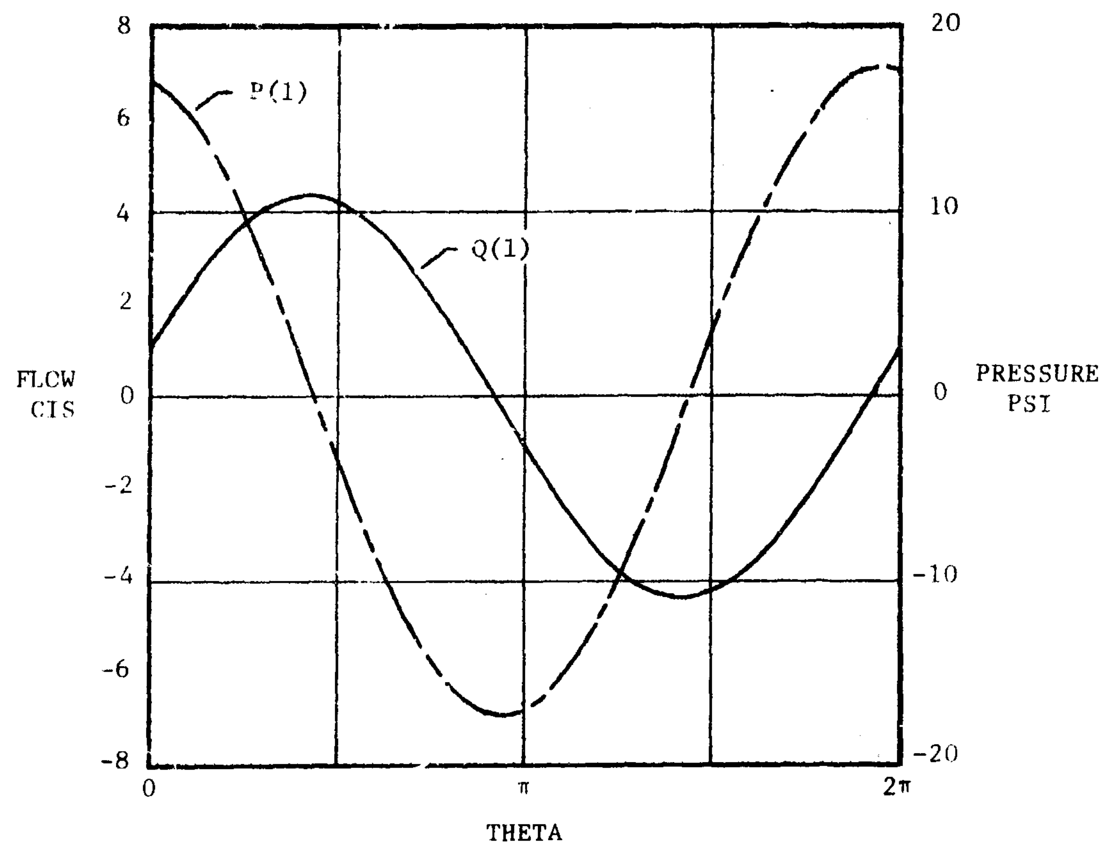


FIGURE 4-13
DYNAMIC FLOW BALANCE

4.8.7 Section 7 - Reconstruct Time Dependent Pressure - Listing

```
C
C      SECTION 7 - RECONSTRUCT TIME DEPENDENT OUTLET PRESSURE
C
  325 CONTINUE
      THECON=2.*PI/80.
      DO 330 J=1,81
      THETA=(J-1)*I*THECON
      PPT(J)=REAL(P3)*COS(THETA)+AIMAG(P3)*SIN(THETA)
      IF(PPT(J).LT.-HPRESS)PPT(J)=-HPRESS
  330 CONTINUE
      IF(IOPT(4).NE.4)GOTO 332
      IF(KK.EQ.1)WRITE(6,321)
  321 FORMAT(//,10X,'**** OUTLET IMPEDANCE VALUES ****',//,
     +5X,'KK',6X,'REAL ZO',5X,'IMAG ZO  REAL ZIP(I) IMAG ZIP(I)',
     +4X,'REAL ZE',5X,'IMAG ZE',/)
      IF(KK.EQ.1)GOTO 332
      WRITE(6,322)KK,ZO,ZIP(I),ZE
  322 FORMAT(3X,I3,2X,6(F10.4,2X))
  332 CONTINUE
      KK=KK+1
      GOTO 170
  335 CONTINUE
C     DO 337 N=1,3
C     WRITE(6,334)N,FQ1(NHARM,N),PQ1(NHARM,N),CABS(FQ1(NHARM,N)),
C    +CABS(PQ1(NHARM,N))
C 334 FORMAT(/,5X,I3,6(2X,F10.4))
C 337 CONTINUE
      Q(1)=0.0001*(1.,0.0)
      IF(QOVB.LE.QMAXIY)Q(1)=PQ1(NHARM,3)/ZIP(NHARM)
      P(1)=0.0001*(0.0,1.)
      IF(QOVB.LE.QMAXIY)P(1)=PQ1(NHARM,3)
      IF(NINLT.EQ.121)GOTO 1900
```

4.9 SECTION 8 - PISTON DECOMPRESSION CALCULATION

The calculation of piston pressures during decompression is identical to the precompression calculation described in Section 3. Valve area index numbers for the decompression portion of cylinder block rotation are used (NDEG = NPRCL+1 TO NSUSOP).

4.9.1 Section 8 - Piston Decompression Calculation - Listing

```
C
C     SECTION 8- PISTON DECOMPRESSION CALCULATION
C
      IF(QOVB.GT.QMAXIY)GOTO 340
      NDEG=NPRCL
      THETA=(NDEG/2.-0.5)/57.3
      XLAST=SSA*SIN(THETA)-SA*COS(THETA)
  360 NDEG=NDEG+1
      THETA=(NDEG/2.-0.5)/57.3
      XNEW=SSA*SIN(THETA)-SA*COS(THETA)
      DVOL=(XNEW-XLAST)*PIA
      VA=POVOL-XLAST*PIA
      NSTART=NDEG-1
      BULKP=BULK+12.*(PISPR(NSTART)-PRESS)
      PISPR(NDEG)=PISPR(NSTART)+BULKP*DVOL/VA
      DLEAK=(PISPR(NDEG)-CPRESS)*PLEAK*DT
      IF(PISPR(NDEG).LT.CPRESS)DLEAK=SQRT(CPRESS-PISPR(NDEG))*
     1SLEAK*DT
      VALEAK=POVOL-XNEW*PIA
      BULKP=BULK+12.*(PISPR(NDEG)-PRESS)
      DPLEAK=BULKP*DLEAK/VALEAK
      PISPR(NDEG)=PISPR(NDEG)-DPLEAK
      IF(PISPR(NDEG).GT.0.01)GOTO 365
      CAVOL=CAVOL-DVOL+DLEAK
      PISPR(NDEG)=0.01
      GOTO 920
  365 CAVOL=0.0
  920 XLAST=XNEW
      IF(IOPT(5).NE.5)GOTO 930
      IF(Y.NE.YOPT(3))GOTO 930
      IF(NDEG.EQ.NPRCL+1)WRITE(6,922)
  922 FORMAT('1',10X,'**** DECOMPRESSION PRESSURES ****',//,
     +5X,'NDEG  THETA     PISTON',8X,'BULK',7X,'CAVOL     PISPR(NDEG)',/,
     +19X,'POSITION  MODULUS',/)
      WRITE(6,924)NDEG,THETA*57.3,XNEW,BULKP,CAVOL,PISPR(NDEG)
  924 FORMAT(5X,I4,2X,F5.1,2X,F10.6,2X,F10.2,2X,F10.6,2X,F10.4)
  930 IF(NDEG.EQ.NSUSOP)GOTO 1000
      GOTO 360
 1000 CONTINUE
      CAVOLD=CAVOL
      DO 1050 N=1,81
      PP1(N)=0.0
 1050 CONTINUE
      KKN=1
      KK=1
 1100 CONTINUE
```

4.10 SECTION 9 - PUMP INLET FLOW CALCULATION

Pump inlet flow is calculated in the same manner as described in Section 4 for outlet flow. Flow values for each piston that is exposed to the suction slot are calculated in 1/2° increments of cylinder block rotation. The flow contribution of all active pistons is summed to define the time dependent inlet flow waveform QQI( ) for the pump.

The inlet code number is tested at the end of Section 9. If inlet analysis is not required (NINLT. NE. 123), control passes to Section 13 where swashplate torques are calculated. If (NINLT. EQ. 123) program execution continues with Section 10.

4.10.1 <u>Section 9 - Pump Inlet Flow Calculation - Listing</u>

```
C
C     SECTION 9- PUMP INLET FLOW CALCULATION
C
      CAVOL=CAVOLD
      DO 1150 N=1,81
      QQI(N)=0.0
 1150 CONTINUE
      NDEG=NSUSOP
      THETA=(NDEG/2.-0.5)/57.3
      XLAST=SSA*SIN(THETA)-SA*COS(THETA)
      NKM=0
      QIN=0.0
      QEST=QIN
      CD=0.65
      NBEG=NSUSOP+1
      DO 1400 NDEG=NBEG,NSUCL
      NKM=NKM+1
      IF(NKM.EQ.81)NKM=1
      THETA=(NDEG/2.-0.5)/57.3
      XNEW=SSA*SIN(THETA)-SA*COS(THETA)
      DX=XNEW-XLAST
      VA=POVOL-XLAST*PIA
      XLAST=XNEW
      QMECH=QMMAX*COS(THETA-XANG)
      IF(CAVOL.GT.0.0)GOTO 1370
      BULKP=BULK+12.*(PISPR(NDEG-1)-PRESS)
      LEAK=SLEAK
      IF(PISPR(NDEG-1).GE.CPRESS)LEAK=PLEAK
      LK1=1./((VA/(BULKP*DT))-LEAK)
      LK2=QMECH+VA*PISPR(NDEG-1)/(BULKP*DT)-CPRESS*LEAK
      LK3=2.*((CD*VAREA(NDEG))**2)/RHO
      QEQUAT=1.
      ITER=1
 1151 CONTINUE
      PTRIAL(ITER)=QEST*ABS(QEST)/LK3+
     1(LPRESS+PPI(NKM))
      QTRIAL(ITER)=LK2-PTRIAL(ITER)/LK1
      IF(ITER.LE.2)GOTO 1152
      IF(ITER.GT.3)GOTO 1153
      IF(QTRIAL(ITER).LE.QTRIAL(ITER-1)
     +.AND.QTRIAL(ITER).GE.QTRIAL(ITER-2)
     +.OR.QTRIAL(ITER).LE.QTRIAL(ITER-2)
     +.AND.QTRIAL(ITER).GE.QTRIAL(ITER-1))GOTO 1153
      QEQUAT=2.
      ITER=1
      QEST=QOUT
 1154 CONTINUE
      PTRIAL(ITER)=LK1*(LK2-QEST)
      QTRIAL(ITER)=SQRT(LK3*ABS(PTRIAL(ITER)-(LPRESS+PPI(NKM))))
      IF(PTRIAL(ITER).LT.(LPRESS+PPI(NKM)))QTRIAL(ITER)=-QTRIAL(ITER)
```

4.10.1 (Continued)

```
      IF(ITER.EQ.1)GOTO 1152
 1153 CONTINUE
      QITERR=ABS(QTRIAL(ITER)-QTRIAL(ITER-1))
      IF(QTRIAL(ITER-1).NE.0.0)QITERR=ABS(QITERR/QTRIAL(ITER-1))
      IF(QITERR.LT.0.005)GOTO 1380
 1152 CONTINUE
      QEST=(QTRIAL(ITER)+QEST)/2.
      IF(ITER.LT.3)QEST=QTRIAL(ITER)
      IF(ITER.EQ.10)GOTO 1380
      ITER=ITER+1
      IF(QEQUAT.EQ.2.)GOTO 1154
      GOTO 1151
 1370 CONTINUE
      LEAK=SLEAK
      QIN=-(CD*VAREA(NDEG)*
     +SQRT(2.*(LPRESS+PPI(NKM))/RHO))
      PISPR(NDEG)=0.01
      CAVOL=CAVOL-DX*PIA+SLEAK*DT*CPRESS+QIN*DT
      IF(CAVOL.LE.0.0)CAVOL=0.0
 1380 CONTINUE
      QIN=QTRIAL(ITER)
      PISPR(NDEG)=PTRIAL(ITER)
      QEST=QIN
      QQI(NKM)=QIN+QQI(NKM)
      IF(IOPT(6).NE.6)GOTO 1400
      IF(Y.NE.YOPT(4))GOTO 1400
      IF(NDEG.EQ.NBEG)WRITE(6,1372)
 1372 FORMAT('1',10X,'**** INLET FLOW VALUES ****',//,
     +5X,'KK NKM NDEG  THETA     PISTON',6X,'PISTON',8X,'QIN',
     +6X,'QQI(NKM)',6X,'CAVOL',5X,'PISPR(NDEG)',/,
     +26X,'POSITION',6X,'FLOW',/)
      WRITE(6,1374)KK,NKM,NDEG,THETA*57.3,XNEW,QMECH,QIN,QQI(NKM),
     +CAVOL,PISPR(NDEG)
 1374 FORMAT(5X,I2,1X,I3,1X,I4,2X,F5.1,2X,F10.6,2X,3(F10.4,2X),F10.6,2X,
     +F10.4)
 1400 CONTINUE
      QQI(81)=QQI(1)
      IF(NINLT.EQ.122) GO TO 350
```

4.11 SECTION 10 - FOURIER ANALYSIS OF PUMP INLET FLOW

The pump inlet flow is mathematically analyzed as described in Section 5 (for outlet flow), except that steady state balancing is not performed.

4.11.1 Section 10 - Fourier Analysis of Pump Inlet Flow - Listing

```
C
C      SECTION 10- FOURIER ANALYSIS OF PUMP INLET FLOW
C
       AN=360./PISTNO
       N=AN+.0001
       COEF=2.0/(2.0*AN+1.0)
       C1=PI*COEF
       S1 =SIN(C1)
       C1 =COS(C1)
       S =0.0
       C =1.0
       FNTZ=QQI(1)
       J =1
 1610  U2=0.0
       U1=0.0
       I=2*N+1
C        FORM FOURIER COEFFICIENTS RECURSIVELY
 1620  U0=QQI(I)+2.0*C*U1-U2
       U2=U1
       U1=U0
       I=I-1
       IF(I-1) 1630,1630,1620
 1630  QIFC(J)=COEF*(FNTZ+C*U1-U2)
       QIFS(J)=COEF*S*U1
       IF(J-NHARM-1) 1640,1650,1650
 1640  U3=C1*C-S1*S
       S =C1*S+S1*C
       C =U3
       J = J+1
       GO TO 1610
 1650  QIFC(1)=QIFC(1)*.5
       IF(IOPT(4).NE.4.OR.KK.NE.1)GOTO 1655
       WRITE(6,1652)QIFC(1)
 1652  FORMAT(//,10X,'*** STEADY STATE INLET FLOW=',F10.4,' CIS ***',//)
C
C        COMPUTE HARMONIC FLOWS FROM FOURIER ANALYSIS
C
 1655  DO 1660 I=1,NHARM
       FQII(I,KK)= CMPLX(QIFC(I+1), QIFS(I+1))
 1660  CONTINUE
```

4.12 SECTION 11 - INLET PRESSURE-FLOW BALANCE CALCULATION

Pump inlet flow is dynamically balanced in the manner described in Section 6.

4.12.1 Section 11 - Inlet Pressure Flow Balance Calculation - Listing

```
C
C     SECTION 11- INLET PRESSURE-FLOW BALANCE CALCULATION
C
 1670 CONTINUE
      I = KKN
      IF (KK - 2) 1680,1690,1700
 1680 CONTINUE
      PQ1I(I,1)=FQ1I(I,1)*ZAP(I)
      P3=PQ1I(I,1)
      GO TO 1720
 1690 CONTINUE
      ZO = PQ1I(I,1) /(FQ1I(I,2)-FQ1I(I,1))
      ZE=ZO*ZAP(I)/(ZO+ZAP(I))
      QSHUNT(1)=PQ1I(I,1)/ZO
      PQ1I(I,2)=PQ1I(I,1)*0.5
      P3=PQ1I(I,2)
      GO TO 1720
 1700 CONTINUE
      J=KKN
      RE=PQ1I(J,2)/(FQ1I(J,3)-FQ1I(J,1))
      ZE=RE*ZAP(J)/(RE+ZAP(J))
      QSHUNT(2)=PQ1I(J,2)/RE
      PQ1I(J,3)=(QSHUNT(1)+QSHUNT(2))*ZO*RE/(ZO+RE)
      IF(IOPT(4).NE.4)GOTO 1710
      WRITE(6,1734)KK,RE,ZAP(J),ZE
 1710 CONTINUE
      KKN = KKN + 1
      IF(KKN.GT.NHARM) GO TO 340
      KK = 1
      FQ1I(KKN,1) = FQ1I(KKN,3)
      GO TO 1670
 1720 CONTINUE
```

4.13 SECTION 12 - RECONSTRUCTION OF TIME DEPENDENT INLET PRESSURE

Pump inlet time dependent pressure is reconstructed as described in Section 7. Balanced inlet flow and pressure are stored for return to the main program.

4.13.1 Section 12 - Reconstruct Time Dependent Inlet Pressure - Listing

```
C
C     SECTION 12- RECONSTRUCTION OF TIME DEPENDENT INLET PRESSURE
C
      DO 1730 J=1,81
      THETA=(J-1)*THECON
      PPI(J) =PPI(J) + REAL(P3)* COS(THETA) +AIMAG(P3)* SIN(THETA)
      IF(PPI(J).LT.-LPRESS) PPI(J)=-LPRESS
 1730 CONTINUE
      IF(IOPT(4).NE.4)GOTO 1735
      IF(KK.EQ.1)WRITE(6,1732)
 1732 FORMAT(//,10X,'**** INLET IMPEDANCE VALUES ****',//,
     +5X,'KK',,6X,'REAL ZO',5X,'IMAG ZO  REAL ZAP(I) IMAG ZAP(I)',
     +4X,'REAL ZE',5X,'IMAG ZE',/)
      IF(KK.EQ.1)GOTO 1735
      WRITE(6,1734)KK,ZO,ZAP(I),ZE
 1734 FORMAT(5X,I2,3X,6(F10.4,2X))
 1735 KK = KK + 1
      GO TO 1100
  340 CONTINUE
      Q(NINLT)=0.0001*(1.,0.0)
      IF(QOVB.LE.QMAXIY)Q(NINLT)=PQ1I(NHARM,3)/ZAP(NHARM)
      P(NINLT)=0.0001*(0.0,1.)
      IF(QOVB.LE.QMAXIY)P(NINLT)=PQ1I(NHARM,3)
  450 CONTINUE
```

4.14 SECTION 13 - TORQUE CALCULATION

This section calculates the average torque acting on the swashplate. These include torques due to spring force, piston mass acceleration forces, and piston pressure forces. Certain pump variables are pre-defined for output plotting, if desired. These include swashangle, pre-compression mismatch, piston pressure at the start of de-compression, cavitation volume, de-compression mismatch, piston pressure at the start of pre-compression, actuator pressure due to spring force, and total actuator pressure.

4.14.1 Math Model - The lever arm (ACTLEV) of the swashplate actuator is estimated based on actual pump geometry as a linear function of swashangle.

$$\text{ACTLEV} = \text{ACTLEVO} + \text{ASWASH}*1.428 \qquad (1)$$

The swashangle is inversely proportion to pump speed for constant flow. This must be changed to suit the particular pump being modeled, since it is not derived from input data.

The swashplate torque due to piston acceleration is the sum of torques due to inertial forces in the variable and steady state motion planes:

$$\text{TORQAC} = \text{Tv} + \text{TRQACR} \qquad (2)$$

$$\text{where Tv} = \text{Fv} * \text{TORQ1} \qquad (3)$$

$$\text{Tv} = \text{M} * \text{a} * \text{TORQ1} \qquad (4)$$

$$\text{Tv} = \text{PIMASS} * \text{a} * \text{TORQ1}. \qquad (5)$$

The piston force moment arm at each increment of piston position (shaft rotation angle) is

$$\text{TORQ1} = \text{RBORC}*\text{Cos(AT)} + \text{HOFF} \qquad (6)$$

For simple harmonic motion, piston acceleration is

$$\text{a} = \text{DY} * \text{w}^2 * \text{Cos(AT)}. \qquad (7)$$

Piston motion maximum amplitude is

$$DY = (RBORC + HOFF)\ (DISAM - DISACT)/ACTLEV \tag{8}$$

for shaft rotation angles from

$$[3\pi/2 - SIN^{-1}\ (HOFF/RBORC)]\ \text{to}\ [\pi/2 + SIN^{-1}\ (HOFF/RBORC)]$$

and $DY = (RBORC - HOFF)\ (DISAM - DISACT)/ACTLEV$

for shaft rotation angles from

$$[\pi/2 + SIN^{-1}\ (HOFF/RBORC)]\ \text{to}\ [3\pi/2 - SIN^{-1}\ (HOFF/RBORC)],$$

where $(DISAM - DISACT)/ACTLEV = TAN(ASWASH)$.

The shaft rotation frequency (W) as a function of pumping frequency is

$$\begin{aligned} w &= Y\pi/30.\ \text{or} \\ w &= W/PISTNO \end{aligned} \tag{9}$$

Inertial piston torque due to the steady state swashplate angle is

$$TRQACR = PIMASS * A_c * TORQ1$$

where $A_c = w^2 * SSA * SIN(AT)$

Therefore,

$$TRQACR = -w^2 * SSA * SIN(AT) * PIMASS * TORQ1. \tag{10}$$

Substituting (41) and (42) into (40), and then into (38) gives

$$\begin{aligned} T_v = &((Y*\pi/30.)**2*(RBORC + HOFF)*(DISAM - DISACT) \\ &*COS(AT)/ACTLEV)*PIMASS*TORQ1 \end{aligned} \tag{11}$$

Substituting (10) and (11) into (2) yields the total piston acceleration torque on the swashplate at any piston position

Swashplate torque due to piston pressure is

$$TORQPR = PISPR(NKM) * PIA * TORQ1.$$

Total swashplate torque due to piston pressure and acceleration is then

$$TORQ = TORQAC + TORQPR$$

The torque at each increment of shaft angle is then summed, divided by the number of calculation points, and multiplied by the number of pistons

to obtain total average torque (TORTAV);

TORQSU = TORQSU + TORQ

TORTAV = TORQSU/ND9 * PISTNO (12)

4.14.2 Assumptions - Swashplate actuator lever arm is assumed to vary linearly with swashangle.

4.14.3 Computation Method - Not applicable.

4.14.4 Approximations - None.

4.14.5 Limitations - Unknown.

4.14.6 Variable Names - See Paragraph 4.1.6.

4.14.7 Section 13 - Torque Calculations - Listing

```
C
C     SECTION 13 - TORQUE CALCULATION
C
      ACTLEV=ACTLEV0+ASWASH*1.428
      TRQACS=0.0
      TORQSU=0.0
      TORQPS=0.0
      TORQAS=0.0
      ATOFF=ASIN(HOFF/RBORC)
      ATPR=0.5*PI+ATOFF
      ATSU=1.5*PI-ATOFF
      DISACT=DISAM-ACTLEV*TAN(ASWASH)
      DO 1810 M=1,NOPIST
      DO 1820 N=1,80
      NKM=NSUCL+N+(M-1)*80
      IF(NKM.GE.ND9+1) NKM=NKM-ND9
      AT=(NKM-1)*AINC/57.3
      TORQ1=RBORC*COS(AT)+HOFF
      TORQPR=PISPR(NKM)*PIA*TORQ1
      TRQACR=-((Y*PI/30.)**2*SSA*SIN(AT)*PIMASS*TORQ1)
      IF(AT.GE.ATSU.OR.AT.LE.ATPR) GO TO 1830
      TORQAC=((Y*PI/30.)**2*(RBORC-HOFF)*(DISAM-DISACT)*
     +COS(AT)/ACTLEV)*PIMASS*TORQ1+TRQACR
      GO TO 1860
 1830 TORQAC=((W/PISTNO)**2*(RBORC+HOFF)*(DISAM-DISACT)*
     +COS(AT)/ACTLEV)*PIMASS*TORQ1+TRQACR
 1860 TORQ=TORQPR+TORQAC
      TORQSU=TORQSU+TORQ
      TORQPS=TORQPS+TORQPR
      TORQAS=TORQAS+TORQAC
      TRQACS=TRQACS+TRQACR
 1820 CONTINUE
 1810 CONTINUE
      TORTAV=TORQSU/ND9*PISTNO
      TORPAV=TORQPS/ND9*PISTNO
      TORAAV=TORQAS/ND9*PISTNO
      TRACAV=TRQACS/ND9*PISTNO
      IF(IOPT(4).NE.4)GOTO 1870
      WRITE(6,1890) TORPAV,TRACAV,TORAAV,TORTAV
 1890 FORMAT(//,10X,'**** TORQUE VALUES ****',//,
     +5X,'AVERAGE TOTAL TORQUE',/,
     +5X,'DUE TO PISTON PRESSURE =',F12.2,' IN-LB',/,
     +5X,'AVERAGE TORQUE DUE TO',/,
     +16X,'CROSS ANGLE =',F12.6,' IN-LB',/,
     +5X,'AVERAGE TOTAL TORQUE',/,
     +6X,'DUE TO PISTON INERTIA =',F12.2,' IN-LB',/,
     +5X,'TOTAL AVERAGE TORQUE    =',F12.2,' IN-LB',/)
 1870 CONTINUE
 1880 Q(37)=CMPLX(ASWASH*57.3,0.0)
      Q(38)=CMPLX((PISPR(NPRSOP)-HPRESS),0.0)
```

4.14.7 <u>(Continued)</u>

```
      Q(39)=CMPLX(PISPR(NPRCL),0.0)
      Q(40)=CMPLX(CAVOLD*1000.,0.0)
      P(37)=CMPLX((PISPR(NSUSOP)-LPRESS),0.0)
      P(39)=CMPLX(PISPR(NSUCL),0.0)
      PSPG=(150.-ASWASH*400.)/ARACT
      PACTU=TORTAV/ACTLEV/ARACT+PSPG+CPRESS
      P(38)=CMPLX(PSPG,0.0)
      P(40)=CMPLX(PACTU,0.0)
 1900 CONTINUE
      RETURN
      END
```

APPENDIX B
OUTPUT REQUIREMENTS
HSFR USER MANUAL (AFAPL-TR-76-43 VOL. III)

2.4 Output Requirements

Input data cards are required to specify desired program output results, which may consist of written and plotted data (at least one data plot must be specified.)

Output write options include listing the following computed data:

(1) pressure and suction slot valve areas

(2) precompression pressure values at a specified RPM

(3) outlet flow calculation values at a specified RPM

(4) flow capability, swash angle/steady state outlet pressure, outlet/ inlet impedance and torque summary values

(5) decompression pressure values at a specified RPM

(6) inlet flow calculation values at a specified RPM

(7) element dynamic flow and pressure values.

Output plot options include:

(1) half-amplitude dynamic flow magnitude vs RPM at specified elements

(2) half-amplitude dynamic pressure magnitude vs RPM at specified elements

(3) half-amplitude impedance magnitude vs RPM at specified elements

(4) acoustic energy density vs RPM

(5) acoustic energy intensity vs RPM

(6) standing pressure half wave amplitude between specified elements of the circuit for up to ten RPM's where peak pressure exceeds a specified test pressure

(7) standing pressure half wave amplitude between specified elements of the circuit at specified RPM's

One card is required to specify the required output write options, and at least one card is required for each output plot option. Each location (element number) in the circuit for which a plot is to be generated must be specified. All cards in the output group must be in the order specified.

One card is required to specify write options. The write option data card is the first card after the circuit element data. Write options are selected by specifying the integer corresponding to the required option in the appropriate data field. If options 2, 3, 5 & 6 are selected, the pump speed for which data is required must be shown in the example card. A blank card (integer zero in the first data field) must be inserted if no write options are required.

At least one card is required for each output plot option except for the standing wave options, as explained in paragraph 2.4.2. The flow output plot option card(s) is the first card after the write output option card. The first integer on the flow output card specifies the total number of flow plots required. The remaining integers identify the circuit elements for which the plots are to be produced. Up to fifteen elements may be listed on this card. If more than fifteen plots are required, the element numbers are specified on the next card, as shown in the example.

Output plot options for pressure, impedance, acoustic energy density and acoustic energy intensity are specified in the same manner as for flow plots. A blank card (integer zero in the first data field) must be inserted, in order, if output plots in a given category are not required.

WRITE OPTION DATA CARD

CARD NUMBER

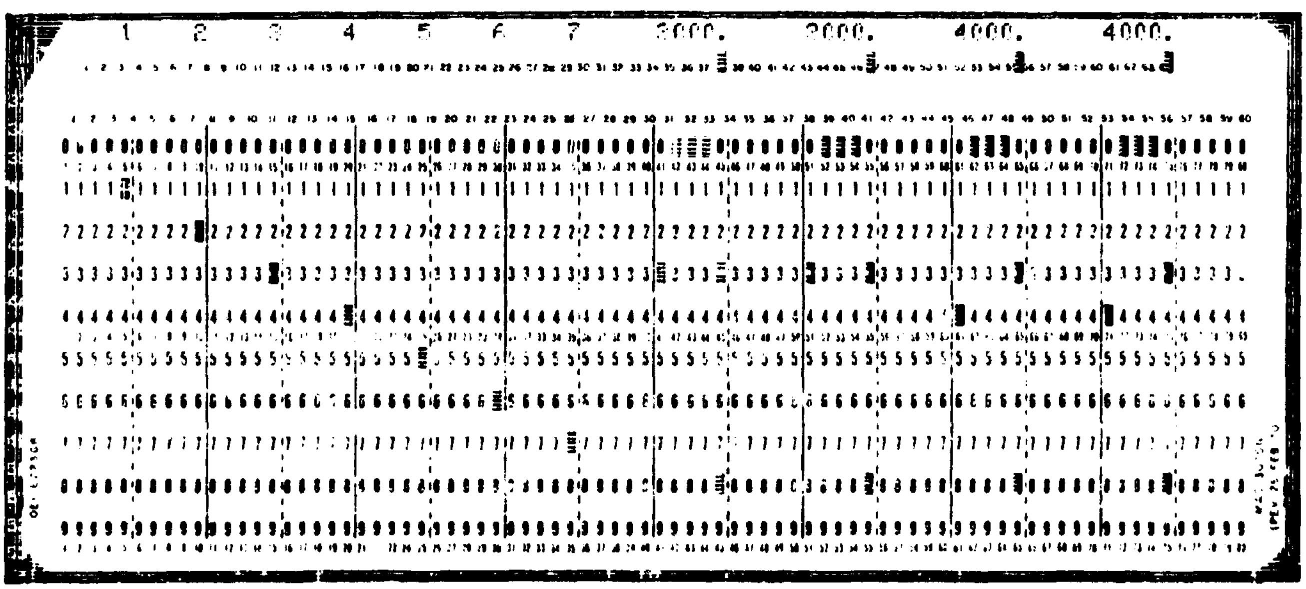
COLUMN	FORMAT	DATA
1-5	I5	1 for Option 1*
6-10	I5	2 for Option 2*
11-15	I5	3 for Option 3*
16-20	I5	4 for Option 4*
21-25	I5	5 for Option 5*
26-30	I5	6 for Option 6*
31-35	I5	7 for Option 7*
36-45	F10.	Speed for Option 2**
46-55	F10.	Speed for Option 3**
56-65	F10.	Speed for Option 5**
66-75	F10.	Speed for Option 6**

* Default value = 0

** Default value = 0.0

NOTE: At least one graph must be selected for plotting.

EXAMPLE CARD

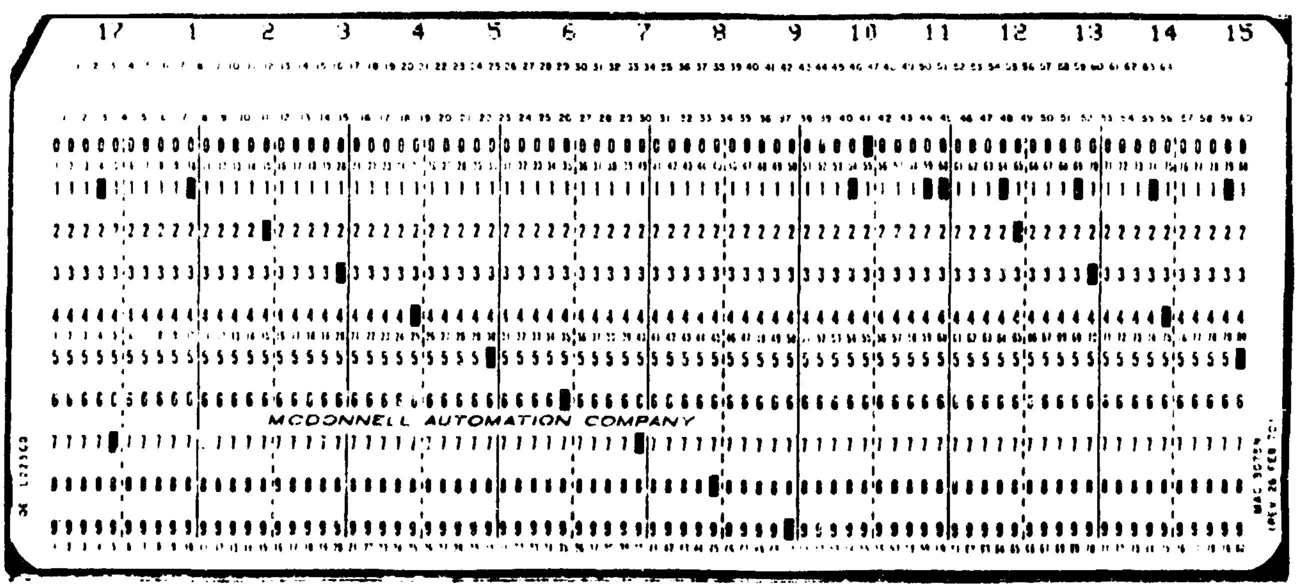


2.4 Output Requirements - Cont'd

First Card

COLUMNS	FORMAT	DATA	DIMENSIONS
1-5	I5	Number of plots for one category	-
6-10	I5	Element number for first plot	-
11-15	I5		
16-20	I5		
21-25	I5		
26-30	I5		
31-35	I5		
36-40	I5		
41-45	I5		
46-50	I5		
51-55	I5		
56-60	I5		
61-65	I5		
66-70	I5		
71-75	I5		
76-80	I5	Element number for fifteenth plot	-

Pressure Plots -- First Card (Typical)

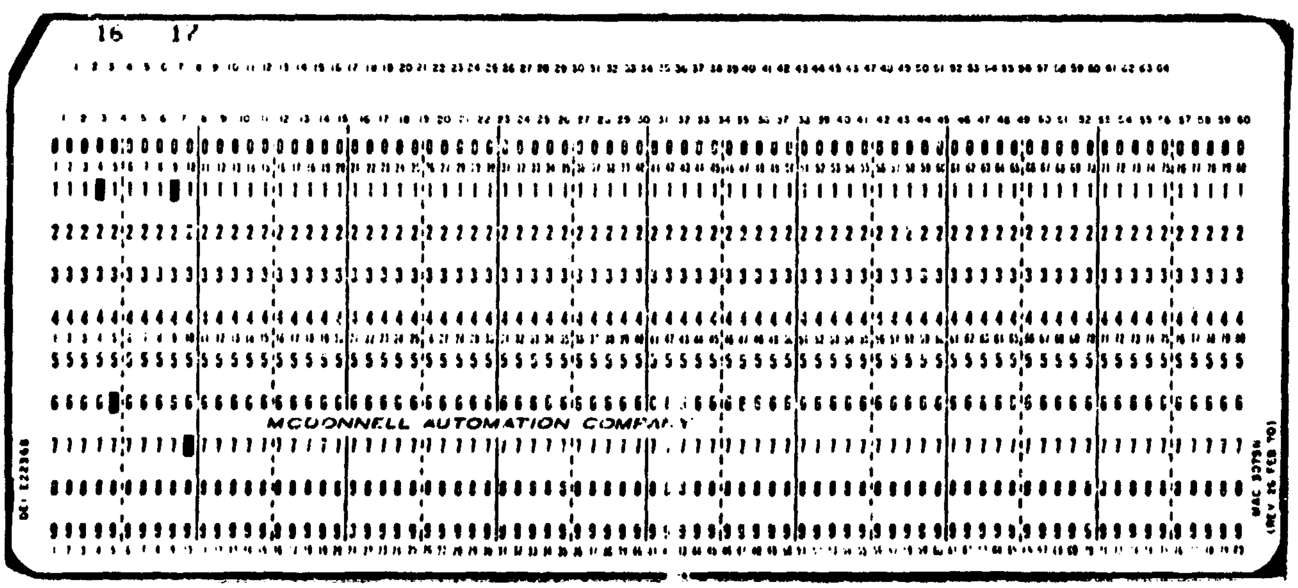


2.4 Output Requirements - Cont'd

Second Card

COLUMNS	FORMAT	DATA	DIMENSIONS
1-5	I5	Element number for sixteenth pressure plot	
6-10	I5	Element number for seventeenth pressure plot	

Pressure Plots - Second Card (Typical)



2.4.1 Pre-defined Output

Any of the normal five plot categories may be used to plot other variables computed within the program. This is useful for studying the behavior of certain variables as a function of pump speed. Normal output titles are retained when making special plots.

Eight variables associated with pump performance are currently pre-defined in the pump subroutine. Any or all of these variables will be plotted if they are selected in the normal manner as described in 2.4 above. These variables have been assigned in the last four positions of the flow (Q) and pressure (P) arrays. The position and description of the pre-defined variables are as follows.

Assigned Name and Location	Description	Units
Q(37)	Swash plate angle	DEGREES
Q(38)	Pre-compressed piston pressure minus steady state outlet pressure (absolute value)	PSI
Q(39)	Piston Pressure at start of decompression	PSI
Q(40)	Piston cavitation at end of decompression	IN^3*1000
P(37)	Decompressed piston pressure minus steady state inlet pressure (absolute valve)	PSI
P(38)	Piston cavitation at start of precompression	IN^3*1000
P(39)	Piston pressure at start of precompression	PSI
P(40)	Swash plate torque due to piston pressure and piston acceleration forces	IN-LBS

The above plot(s), if selected for output, will plot regardless of the number of circuit elements in the model.

2.4.2 STANDING WAVE PLOT OUTPUT REQUIREMENTS

The standing wave plot feature provides two options as specified on card number three of the HSFR data deck. Option one plots standing waves at RPM's where the peak pressure value exceeds a user specified maximum pressure within a bandwidth. The second option allows the programmer to obtain a standing wave plot at a selected RPM.

The standing wave output requirement cards immediately follow the acoustic intensity plot card(s). One card is required for the first standing wave option. The card contains a series of integer values. The first value gives the total number of element pairs times two. The remaining integers are entered in pairs and specify the beginning and ending elements for the section of the system in which the standing waves are to be plotted. (The ending element need not be a HSFR terminating element type).

When a section of the system contains branched lines, only the main section will be plotted, provided the system elements are numbered according to the scheme shown in Figure 2-1A. The user must separately select each branch to be plotted.

The HSFR program will print standing wave plots for ten RPM's. After ten a message will be printed indicating the unplotted resonant RPM's.

The second option, which requires two data cards, allows the programmer to select an RPM for plotting a standing wave. On the first card the user chooses the sections by specifying the beginning and ending element numbers. The second card contains the RPM(s) to be plotted.

Each option can be selected separately, but option one must be chosen before option two.

The HSFR program will output standing wave plots at RPM's where the computed peak pressure exceeds a specified maximum pressure (option one) and at user selected RPM's (option two). The plots will be in the form of peak pressure versus distance.

In columns 51-60 of card three, the user can specify the maximum standing wave pressure. If this value is not specified, the program will not provide standing wave plots at computed peak pressures. The bandwidth for the peak pressure is input in the next data field. One hundred RPM is the default bandwidth value. The program will plot at the RPM of the maximum pressure in the bandwidth.

To output a standing wave at a selected RPM, a real number is placed in columns 71-80. The standing wave maximum pressure need not be specified for this option.

If any of the other plotting options (flow, pressure, impedance, acoustic density, or acoustic intensity) are not selected by the user, blank cards must be inserted in the output cards where they would normally be specified.

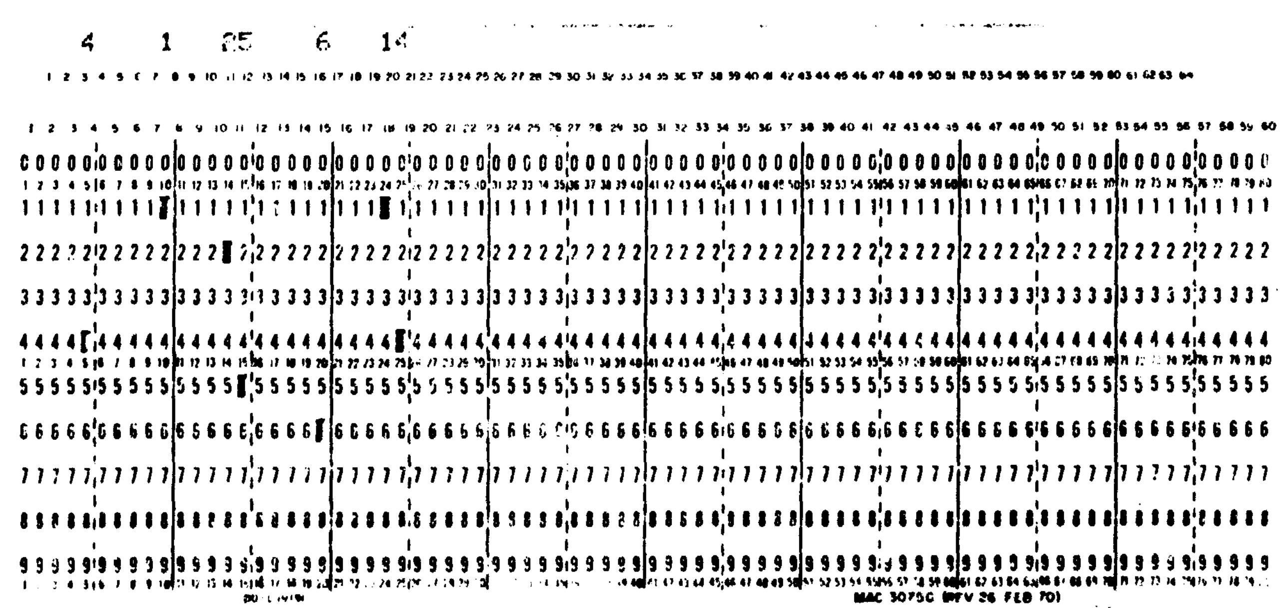
Section 2.4.2 contains further data card requirements that must be entered to obtain the standing wave plots.

OUTPUT REQUIREMENTS - OPTION ONE - COMPUTER GENERATED PLOTS OVER

SPECIFIED MAXIMUM PRESSURE

COLUMNS	FORMAT	DATA	DIMENSIONS
1-5	I5	Number of Element Pairs Times Two	-
6-10	I5	First Element Number	-
11-15	I5	Last Element Number	-
16-20	I5	First Element Number	-
21-25	I5	Last Element Number	-
26-30	I5	Repeat in pairs for up to seven sections	
31-35	I5	of the modeled system	
36-40	I5		
41-45	I5		
46-50	I5		
51-55	I5		
56-60	I5		
61-65	I5		
66-70	I5		
71-75	I5		
75-80	I5		

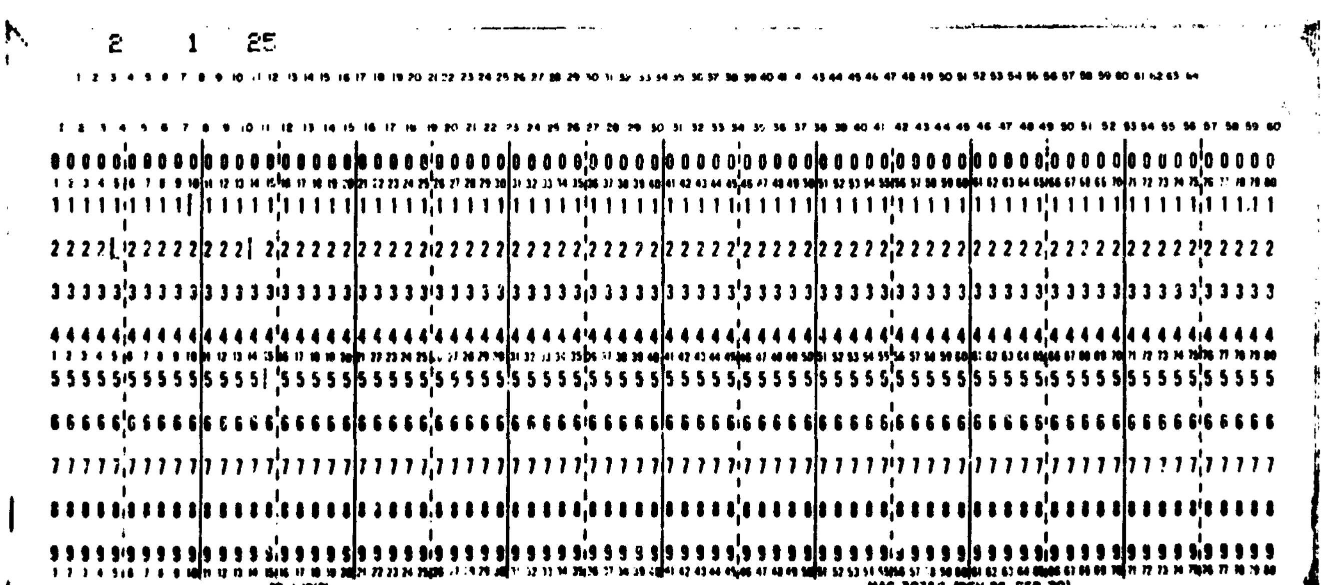
OPTION ONE-STANDING WAVE PLOTS - (TYPICAL DATA CARD)



OUTPUT REQUIREMENTS - OPTION TWO - USER SELECTED RPM

COLUMNS	FORMAT	DATA	DIMENSIONS
1-5	I5	Number of Element Pairs Time Two	-
6-10	I5	First Element Number	-
11-15	I5	Last Element Number	-
16-20	I5	Repeat in pairs for up	
21-25	I5	to seven sections	
26-30	I5	of the modeled system	
31-35	I5		
36-40	I5		
41-45	I5		
46-50	I5		
51-55	I5		
56-60	I5		
61-65	I5		
66-70	I5		
71-75	I5		
76-80	I5		

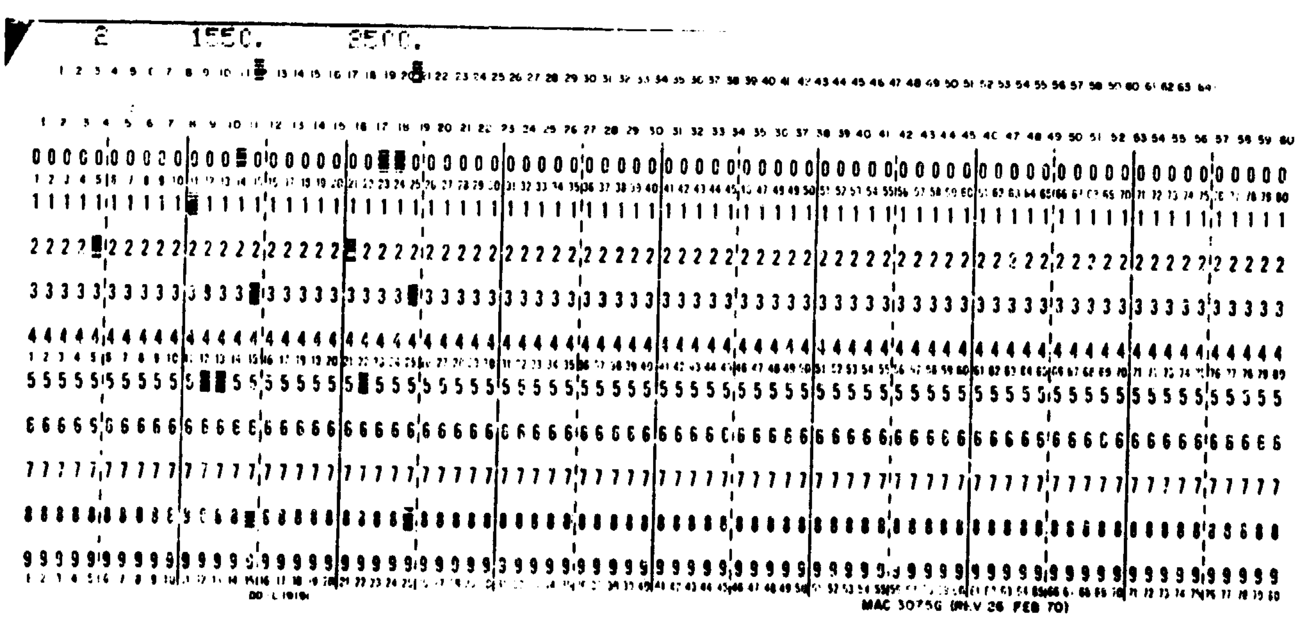
OPTION TWO STANDING WAVE PLOTS - (TYPICAL DATA CARD)



OUTPUT REQUIREMENTS - OPTION TWO - USER SELECTED RPM

COLUMN	FORMAT	DATA	DIMENSIONS
1-5	I5	Number of RPM's	-
6-15	E10.0	Speed for First Standing Wave Plot	RPM
16-25	E10.0	Speed for Second Standing Wave Plot	RPM
26-35	E10.0	" " " " " "	
36-45	E10.0		
46-55	E10.0		
56-65	E10.0		
66-75	E10.0	Speed for Seventh Standing Wave Plot (7 Max)	RPM

OPTION TWO STANDING WAVE PLOTS - (TYPICAL DATA CARD)



2.5 Program Deck and Input Data Job Set-Up

The program deck and input data cards are assembled as shown in Figure 2-6. All input data cards must be in the order shown. Circuit element data cards must be sequential starting with element number 1 and ending with the last, highest number terminated element. Job control card content and arrangement varies depending on the particular computer facility available to the user.

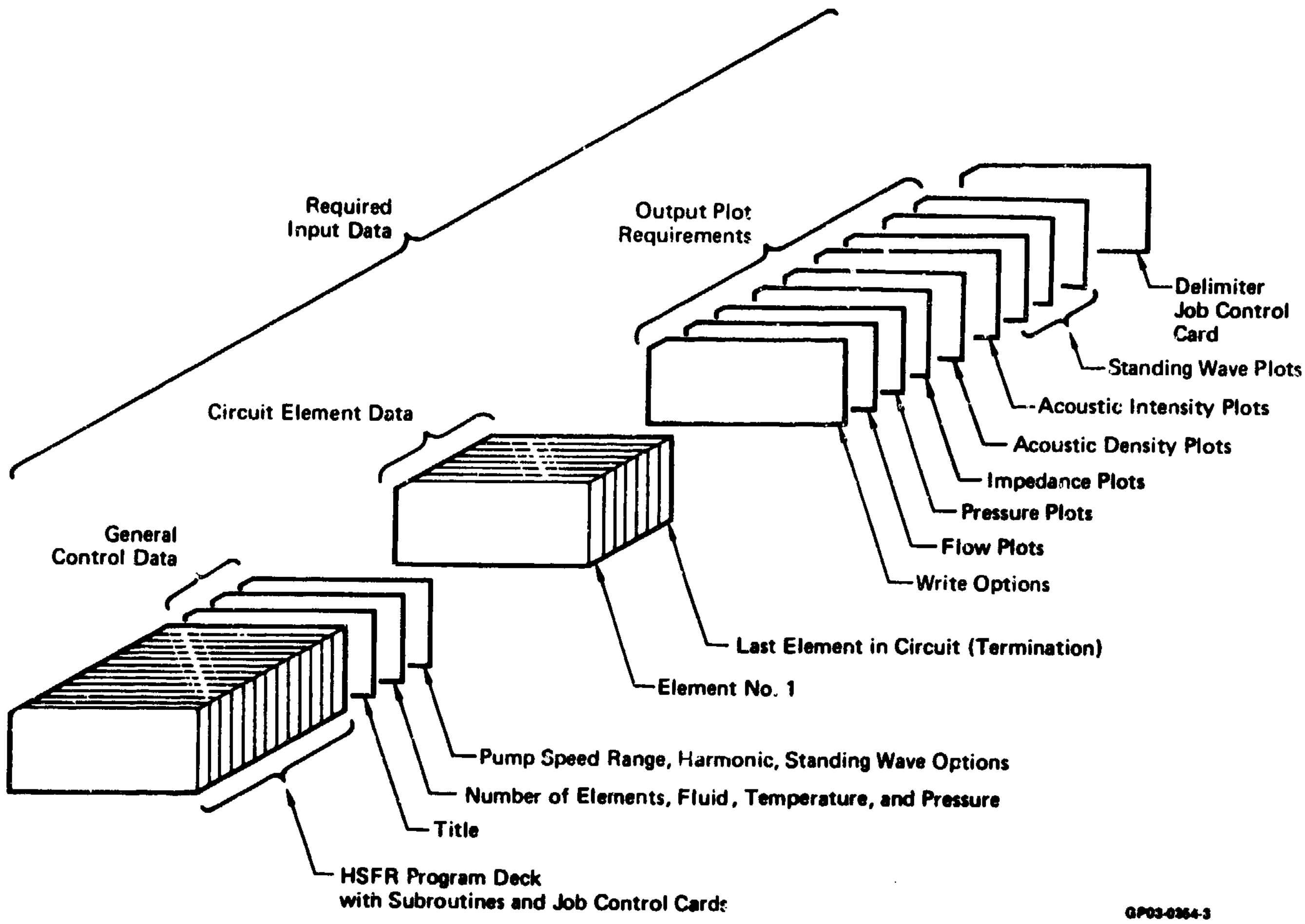


FIGURE 2-6
HSFR PROGRAM DECK AND INPUT DATA JOB SETUP

3.0 PROGRAM OUTPUT

3.1 Computer Output

Figure 3-1 shows an example of the first information printed by the program for a given short line test circuit.

The program prefaces the listing of input data with the program name, date of run, run title, pump speed range and increment, harmonic, and a list of the fluid properties calculated by the program for the specified fluid, temperature and pressure. A list of the input data for all of the elements used in the run is printed next. Finally, the program prints the total number of flow, pressure, impedance, acoustic energy density, and acoustic energy intensity plots required, and, in each case, the specified circuit element numbers for which plots are desired.

Computed output data is printed after the listing of circuit input and plot requirement information provided that any of the seven output write options discussed in Section 2.4 are selected. An example of write option 4 and 7 output is shown in Figure 3-2 for the sample test circuit at 1000 RPM. Extra WRITE statements may be incorporated so that any parameter contained in the program is printed. Extra WRITE statements may be useful to investigate the behavior of variables of interest.

Finally, the required output plots are printed.

3.2 Output Plots

The computed program output normally consists of printed plots of flow, pressure, impedance, and acoustic energy density and intensity versus pump speed for selected junctions in the circuit being analyzed. Additionally, the program provides for printing plots of the standing pressure wave in selected sections of the circuit. The manner in which the user specifies the required plots is described in Section 2.4. The functional subroutine GRAPH2, which is used to produce the plots, is described in Volume IV. The restrictions placed

HYDRAULIC SYSTEM FREQUENCY RESPONSE PROGRAM

**** HSFR - SHORT LINE TEST CIRCUIT - 0.0 GPM - (OSLTC) ****

RESPONSE IS CALCULATED FROM 1000.00 TO 5000.00 R.P.M. IN INCREMENTS OF 50.00 R.P.M.

RESPONSE IS PLOTTED FOR THE FIRST HARMONIC FREQUENCY ,

NUMBER OF PUMPING PISTONS= 9.

FLUID DATA FOR MIL-H-5606A AT 3000.0 PSIG AND 135.0 DEG F

VISCOSITY = .174E-01 IN**2/SEC
DENSITY = .811E-04 (LB-SEC**2)/IN**4
BULK MODULUS = .219E+06 PSI

ELEMENT NUMBER ********************SYSTEM ELEMENT INPUT DATA********************

ELEMENT NUMBER	N TYPE	K TYPE	........PHYSICAL DATA........							
1	9	21	.190	.666	1.120	1.172	.698	.570	.180	
			.20000	19.50000	3.60000	3.37500	28.75000	26.25000	26.00000	21.75000
			50.00000	.06000	.78000	2.07000	.00042	55.00000	150.00000	.67000
2	3	0	4.000	0.030	0.000	0.000	0.000	0.000	0.000	
3	1	0	8.000	1.200	.100	30000000.000	0.000	0.000	0.000	
4	1	0	3.750	1.000	.058	30000000.000	0.000	0.000	0.000	
5	1	0	2.870	1.000	.058	30000000.000	0.000	0.000	0.000	
6	6	1	0.000	0.000	0.000	0.000	0.000	0.000	0.000	
7	13	0	.150	0.000	0.000	0.000	0.000	0.000	0.000	
8	1	0	1.380	1.000	.058	30000000.000	0.000	0.000	0.000	
9	1	0	5.000	1.000	.058	30000000.000	0.000	0.000	0.000	
10	1	0	5.000	1.000	.058	30000000.000	0.000	0.000	0.000	
11	1	0	5.000	1.000	.058	30000000.000	0.000	0.000	0.000	
12	1	0	5.000	1.000	.058	30000000.000	0.000	0.000	0.000	
13	1	0	54.620	1.000	.058	30000000.000	0.000	0.000	0.000	
14	14	0	780.000	0.000	0.000	0.000	0.000	0.000	0.000	

1 Q PLOTS FOR INPUT TO ELEMENT NUMBERS 1
7 P PLOTS FOR INPUT TO ELEMENT NUMBERS 1 5 9 10 11 12 13

FIGURE 3-1 TYPICAL PRE-PLOT OUTPUT

** DATA FOR 1000. RPM

*** SWASH ANGLE-STEADY STATE OUTLET PRESSURE CALCULATION ***

IFL	SWASH ANGLE	ST.STATE FLOW	FLOW ERROR	ST.STATE PRESSURE
1	1.5107	-.3604	-.3604	3000.0000
2	1.6619	-.0121	-.0121	3000.0000
3	1.6670	-.0004	-.0004	3000.0000

**** FLOW CALCULATION DATA ****

SWASH ANGLE : 1.6670 DEG
FLOW DEMAND : 0.0000 CIS
FLOW CAPABILITY
AT 3000. PSI : 42.7011 CIS
AT ZERO PSI : 46.1532 CIS

**** OUTLET IMPEDANCE VALUES ****

KK	REAL ZO	IMAG ZO	REAL ZIP(I)	IMAG ZIP(I)	REAL ZE	IMAG ZE
2	-20.0126	-103.8769	.3955	6.1913	.3652	6.5757
3	-20.0116	-103.8000	.3955	6.1913	.3652	6.5757

**** ELEMENT DATA ****

I	REAL Q(I)	IMAG Q(I)	REAL P(I)	IMAG P(I)	CABS(Q(I))	CABS(P(I))
1	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
2	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
3	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
4	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
5	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
6	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
7	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
8	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
9	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
10	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
11	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
12	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
13	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
14	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]

FIGURE 3-2 WRITE OPTION DATA OUTPUT

on GRAPH2 in the specified pump speeds are given in Section 2.2.

3.3 Interpretation of Computer Output Plots

A typical print-plot output of the first harmonic (fundamental) frequency amplitude of the dynamic peak pressure versus pump speed is shown in Figure 3-3. This is a plot of pressure data computed for the junction of element numbers 10 and 11 in the sample short line test circuit. The computed points are shown superimposed on actual test data for that location in Figure 3-4.

The predicted resonant frequencies occur at about 1300 RPM, 2500-2600 RPM, and 3900 RPM. These appear to correlate well with the measured data except for the first peak. This discrepancy is believed to result from an effect of pump compensator dynamics at low pump speed which is not accounted for in the program.

3.3.1 RPM Bandwidth Sensitivity - Note that the amplitude of the apparent predicted peak pressure at the major resonance (3900 RPM) for this location is significantly lower than the measured data. Consider that the predicted values are calculated at 50 RPM increments of pump speed and that the bandwidth of the predicted peak is quite narrow; less than 200 RPM for peak pressures greater than 100 psip.

Figure 3-5 shows predicted data for 5 RPM increments of pump speed for the speed range around the major resonant frequency. Note that the correlation with test data is better; improving from 385 psip (using 50 RPM increments) to 437 psip. The amplitude correlation error is reduced from about -20% to less than -10%.

The 50 RPM pump speed increment may be used over a wide range of pump speeds (1000-5000 RPM) to identify the major resonant frequencies. However, it is recommended that a smaller incremental speed (5-10 RPM) be used for analysis around the predicted resonant frequency to assure that the peak pressure is identified. This is especially important when the bandwidths of the resonant peak is narrow.

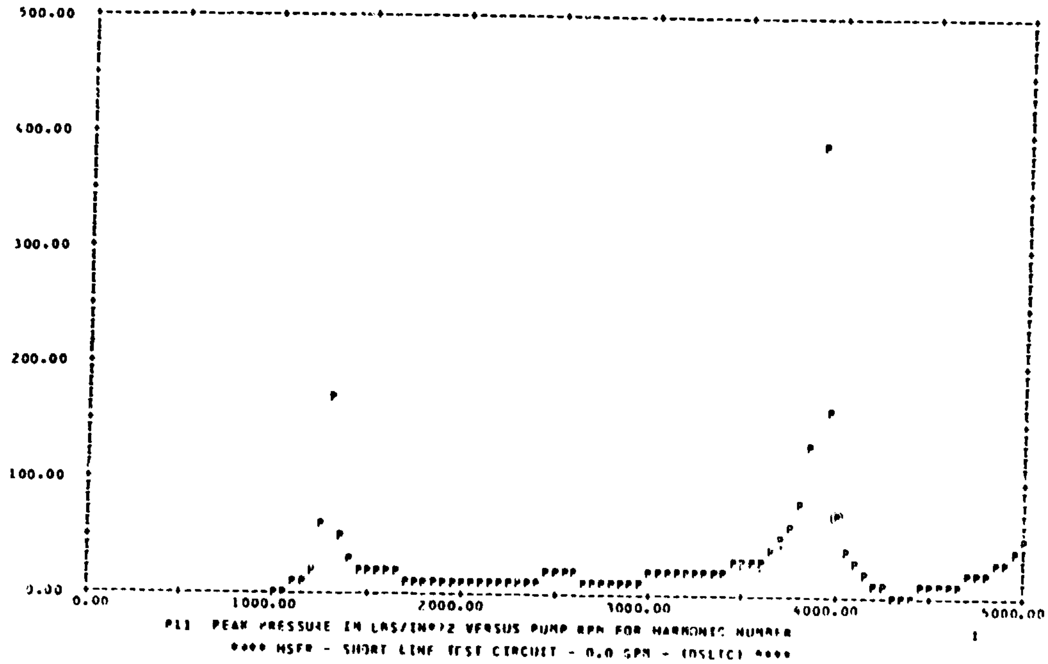
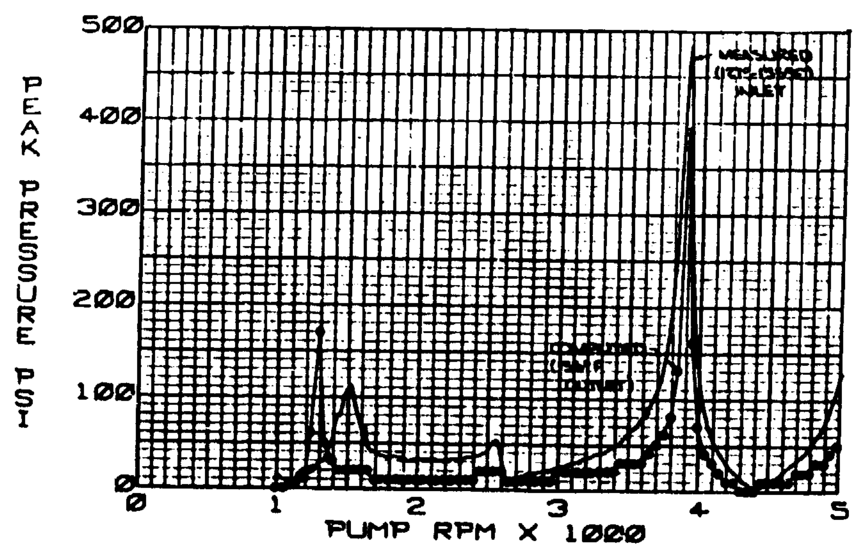


FIGURE 3-3 TYPICAL PRESSURE PRINT-PLOT



F-15 HYDRAULIC PUMP
64-83-P1 FUNDAMENTAL FREQ.
Ø CIS 13Ø DEG F

FIGURE 3-4 DATA CORRELATION

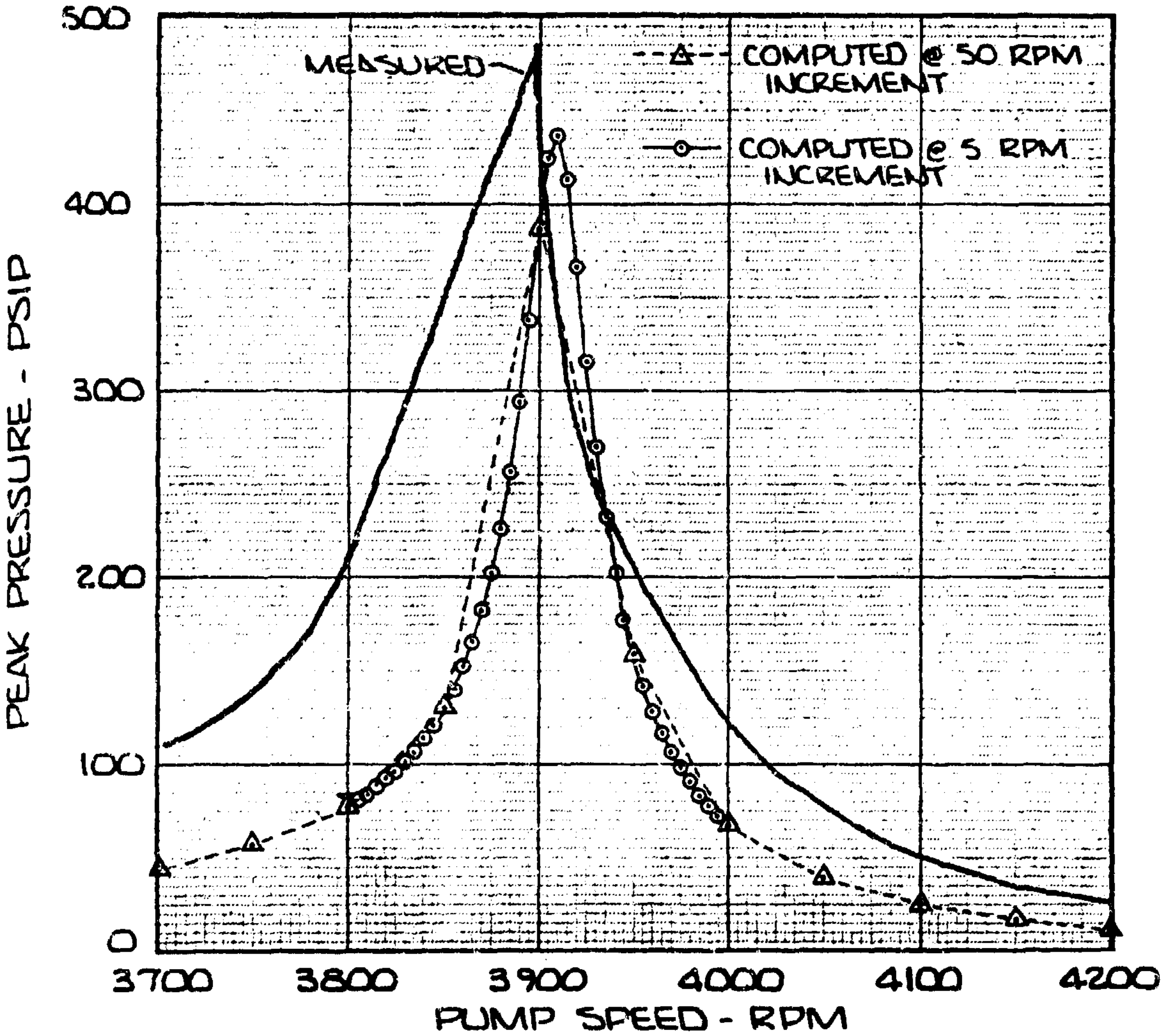


FIGURE 3-5
PRESSURE FREQUENCY RESPONSE
RPM BANDWIDTH SENSITIVITY

The results obtained by the foregoing analaysis are considered adequate to establish the pressure frequency response characteristic of the system being evaluated. Attenuation techniques may be applied to relocate the resonant frequency or reduce the amplitude to acceptable levels.

3.3.2 Temperature-Position Sensitivity - The input data used to define the system includes specifying the fluid temperature and the locations within the circuit for which analysis is desired. The sensitivity of the calculated results to these parameters is illustrated in Figure 3-6.

Calculated pressure frequency response data are shown for three points in the example circuit with 5 inch spacing at 130°F, 135°F, and 150°F fluid temperatures. Point P11 is the location for which test data was recorded. The fluid temperature varied from 127°F to 135°F reported at the inlet to the pump.

Peak pressure varies more than 300 psi within ± 5 inches of point P11. This is explained by the characteristics of the traveling and standing pressure waves that exist in a closed system at each harmonic frequency. Appendix A of Volume IV describes how these waves are distributed, and the relationship between pressure and flow with respect to distance along the line. Care must be taken to avoid drawing erroneous conclusions from the results of a single point (or widely spaced points) pressure measurement or calculation.

Figure 3-6 also shows sensitivity to temperature. Note that the resonant frequency pump speed shifts approximately -100 RPM for a +20°F temperature change within the range of temperatures being evaluated. While the downstream fluid temperatures were not recorded for the test data shown, experience with the F-15 pump indicates that a 10°-15°F temperature rise across the pump may be expected. Therefore, the actual fluid temperature at which data were recorded most likely was in the 135°-150°F range.

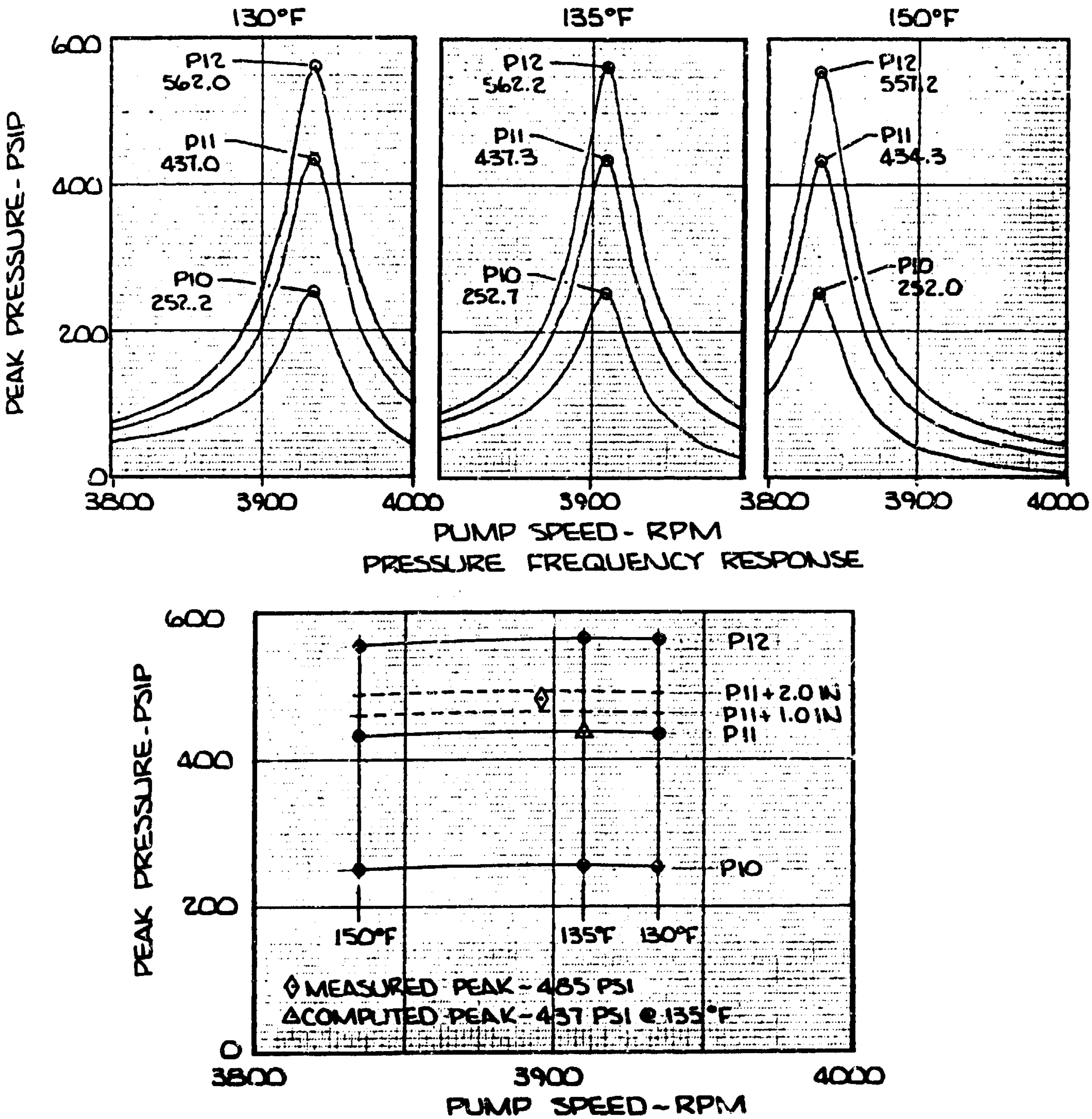


FIGURE 3-6
TEMPERATURE-POSITION SENSITIVITY

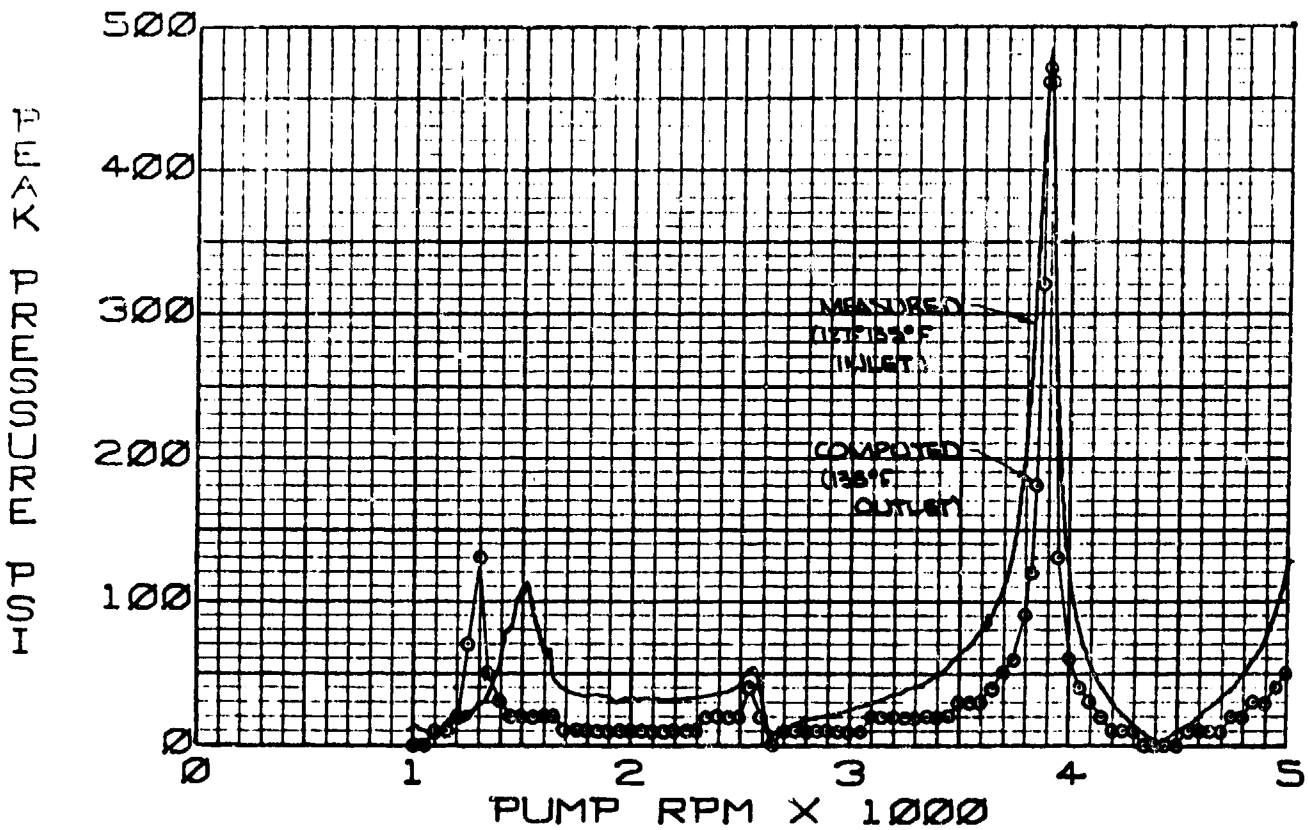
The peak pressure values, location, and temperature data from the pressure frequency response plots were used to construct the temperature - position sensitivity envelope shown in Figure 3-6. The measured peak pressure (from test data) is shown, along with the calculated peak pressure at the supposed condition of 135°F at location P11.

The sensitivity relationship shown in this envelope suggests two points for consideration: (a) a fluid temperature of 138°F would provide more exact resonant frequency correlation; and, (b) a location approximately 1.5 inches farther downstream from P11 would correlate with the measured pressure amplitude.

Accordingly, the program was rerun with these adjustments to the input data. Speed increments of 50 RPM were used for the range of 1000-5000 RPM and 5 RPM increments were used between 3800 and 4000 RPM to define the resonant pressure peak. The results are presented in Figure 3-7. Excellent pressure amplitude correlation at the 3900 RPM resonant speed exists. Note also that better amplitude correlation results at about 2550 RPM, (using the 50 RPM speed increment) compared to the calculated data shown in Figure 3-4.

3.3.3 Discussion of Sensitivity Analysis - The adjustments to input data that were made to achieve correlation are considered well within reason for this analysis. The original data is from F-15 pump verification tests that were conducted in 1976. Exact positional measurements are no longer available, however, the 1.5 inch difference required for exact correlation can be accounted for in the fittings used to couple the short line test circuit.

The sensitivity analysis emphasizes the need for rigidly exact definition of the circuit and precise temperature measurement when single point correlation work is being performed. For the purpose of system analsyis, the sensitivity of output data to input parameters suggests that a number of fairly closely spaced calcuation points in a well defined system need to be used.



F-15 HYDRAULIC PUMP
64-83-P1 FUNDAMENTAL FREQ.
0 CIS 130 DEG F

FIGURE 3-7 DATA CORRELATION

Wide band (50 RPM) speed increments may be used to establish the approximate resonant frequency speeds, but narrow band (5-10 RPM) speed increments are required to accurately define peak pressure amplitudes. Standing wave analysis is definitely recommended, using either the programmed print-plot option or by manually plotting computed points.

APPENDIX B
STANDING WAVE PLOT OPTION
HSFR TECHNICAL MANUAL (AFAPL-TR-76-43, VOL IV)

SECTION 7 - STANDING WAVE PLOTS

The programmer selects the HSFR standing wave plotting capability as described in the user manual. If the maximum standing wave pressure (SWPR) is greater than zero, the beginning and ending element numbers of the chosen section are read.

During program execution peak pressures for each element are stored in SWPR( ) versus pump RPM. If any element pressure value exceeds SWMP the RPM is saved in SWRPM (NSWP) and the corresponding column number of SWPR( ) is placed in ICOL(NSWP). (NSWP is the counter for standing wave pressure plots.) The pressure is written into SWRPMP (NSWP) where it is compared to other resonant pressures in the specified bandwidth. The maximum resonant pressure is chosen for plotting.

If the current resonant pressure is greater than the previous resonant pressure in the selected bandwidth, the program will assign the current pressure as the main resonance RPM. When the pressure fails by more than 10 PSI from the previous value, a new resonant RPM is not assigned.

Should there be no pressure above the maximum standing wave pressure no standing wave will be plotted and a message will be printed. If the number of standing wave plots exceeds ten, they will not be plotted, but the RPM at which a resonance occurs will be printed.

Once the column numbers and hence the RPM's of the SWPR( ) array that contain resonant pressures are known, the length of each plotted section is computed and stored in OMEGA( ). A call to GRAPH2 plots the peak pressure versus distance.

If PRPM and SWMP are both greater than zero, the user has specified an RPM to plot the standing wave. The beginning and ending element numbers and the RPM's are read. The program locates the proper columns in the SWPR( ) array. The pressures in these columns are plotted versus distance with the same call to GRAPH2. Write statements provide the resonant RPM, and the beginning and ending element numbers of the selected branch to be plotted.

VARIABLE LISTING FOR STANDING WAVE
SECTION IN HSFR

NAME	DESCRIPTION	DIMENSION
BWRPM	Resonant Bandwidth	RPM
IELM( )	Element Numbers in the Standing Wave Section	--
ISWEL( )	Array Containing Beginning and Ending Element Numbers for the Standing Wave Section	--
ICOL	Column Locations of Resonant RPM's in SWPR( ) Array	--
IXAX	Number of Elements in Standing Wave Section	--
NSWP	Number of Standing Wave Plots	--
NELS	Start Element Number for Standing Wave Plot	--
NELE	End Element Number for Standing Wave Plot	--
SWMP	Standing Wave Maximum Pressure	PSIP
PRPM	Indicator for Plotting Standing Wave at User Selected RPM	--
SWPR( )	Component Peak Pressure Array Stored Versus RPM	PSIP
SWRPM( )	Array of Standing Wave RPM's	RPM
SWRPMP( )	Array of Standing Wave RPM Pressures	PSIP
ALEN	Length of Standing Wave Section	IN

```
C
C *SECTION 2- COMPUTE TRANSFER MATRIX FOR EACH CIRCUIT ELEMENT*
C
      IF(NHARM.EQ.0) NHARM=1
  110 CONTINUE
      KHARM=0
      IF(KTYPE(1).EQ.21)NINLT=121
      IF(KTYPE(1).EQ.22) NINLT=122
  120 CONTINUE
      QOVB=0.0
      FLOW=0.0
      KHARM=KHARM+1
      A=W*PISTNO*PI*KHARM/30.
      DO 250 IEL=1,NEL
      ITYPE=NTYPE(IEL)
      GO TO (130,140,150,160,170,180,175,190,200,250,130,140,150,160,
     +170,180,175,190),ITYPE
  130 CALL LINE
      GO TO 250
```

```
C *SECTION 7- STANDING WAVE PLOTTING
C
      IF(SWMP.LE.0.0)GO TO 1150
      IWPT=IWPT-1
C     DO 900 I=1,NEL
C     WRITE(6,9011)(SWPR(I,J),J=1,IWPT)
C 900 CONTINUE
 9011 FORMAT(//,10(5X,10E12.5,/))
 1140 CONTINUE
      NBRAN=NBRAN/2
      DO 2000 M=1,NBRAN
      NELS=ISWEL(2*M-1)
      NELE=ISWEL(2*M)
      IF(PRPM.GT.0.0.AND.SWMP.LE.0.0)GO TO 1100
      NSWP=0
      W=WSTART
      DO 1010 I=1,IWPT
      DO 1015 I1=NELS,NELE
      IF(SWPR(I1,I).GE.SWMP)GO TO 1016
 1015 CONTINUE
      GO TO 1050
 1016 NSWP=NSWP+1
      IF(NSWP.GT.10)GO TO 1030
      SWRPMP(NSWP)=SWPR(I1,I)
      SWRPM(NSWP)=W
      ICOL(NSWP)=I
      IF(NSWP.EQ.1)GO TO 1050
C     TEST IF IN SPECIFIED BANDWIDTH
      IF((SWRPM(NSWP-1)+BWRPM).GT.W)GO TO 1020
      GO TO 1050
 1020 CONTINUE
      IF(SWPR(I1,I).GT.SWRPMP(NSWP-1))GO TO 1040
      NSWP=NSWP-1
      GO TO 1050
 1030 WRITE(6,9000)W
 9000 FORMAT(//,10X,45H***** THERE IS AN UNPLOTTED STANDING WAVE AT ,
     + 1F12.4,7H R.P.M.)
      GO TO 1050
 1040 SWRPM(NSWP-1)=W
      ICOL(NSWP-1)=I
      SWRPMP(NSWP-1)=SWRPMP(NSWP)
      NSWP=NSWP-1
C     CALCULATE RPM
 1050 W=W+WINC
 1010 CONTINUE
C     WRITE(6,9020)NEL,IWPT,NSWP,NELS,NELE
 9020 FORMAT(/,20X,'*** INTEGER INFO ',6I10)
      IF(NSWP.EQ.0)GO TO 100
C     WRITE(6,9011)(SWRPM(I),I=1,NSWP)
C     WRITE(6,9021)(ICOL(I),I=1,NSWP)
C     WRITE(6,9011)(SWRPMP(I),I=1,NSWP)
      GO TO 1120
```

```
 1100 DO 1110 I=1,NSWP
 1110 ICOL(I)=(SWRPM(I)-WSTART)/WINC+1
 1120 CONTINUE
      IXAX=1
      ALEN=0.0
      OMEGA(1)=0.0
      I=NELS-1
 1200 CONTINUE
      I=I+1
      IF(I.GT.NELE)GO TO 1300
      IF(NTYPE(I).EQ.1)GO TO 1210
      IF(NTYPE(I).EQ.6.AND.I.NE.NELS)GO TO 1240
      IF(NTYPE(I).EQ.2.AND.KTYPE(I).EQ.32)GO TO 1250
      IF(NTYPE(I)-10)1260,1200,1220
 1250 KK=PARM(8,I)
      ALEN=ALEN+PARM(1,KK)
      GO TO 1230
 1220 IELM(IXAX)=I
      GO TO 1200
 1210 IELM(IXAX)=I
      ALEN=ALEN+PARM(1,I)
 1230 IXAX=IXAX+1
      OMEGA(IXAX)=ALEN
      GO TO 1200
C     SKIPS BRANCH CIRCUIT ELEMENTS
 1240 I=KTYPE(I)+I
      GO TO 1200
 1260 IF(I.EQ.NELE)GO TO 1220
      GO TO 1200
 1300 CONTINUE
C     WRITE(6,9020)IXAX,NEL,NSWP,NELS,NELE,NBRAN
C     WRITE(6,9011)(SWRPM(I),I=1,NSWP)
C     WRITE(6,9021)(NTYPE(I),I=NELS,NELE)
C     WRITE(6,9021)(IELM(I),I=1,IXAX)
C     WRITE(6,9021)(ICOL(I),I=1,NSWP)
 9021 FORMAT(//,10(10X,10I10,/))
C     WRITE(6,9011)(OMEGA(I),I=1,IXAX)
      CHAR(1)=PC
      DO 1400 I=1,NSWP
      DO 1600 J1=1,IXAX
 1600 YPLT(J1)=SWPR(IELM(J1),ICOL(I))
      CALL GRAPH2(OMEGA,YPLT,IXAX,CHAR)
      WRITE(6,9010)NHARM,SWRPM(I)
 9010 FORMAT(/,34X,40HPEAK PRESSURE (LB/IN**2) VERSUS DISTANCE,
     + 26H (IN) FOR HARMONIC NUMBER ,I3,2X,4H AT ,F10.2,7H R.P.M.)
      WRITE(6,9012)NELS,NELE
 9012 FORMAT(/,40X,41HTHE STANDING WAVE IS PLOTTED FROM ELEMENT,
     + I5,14H   TO ELEMENT ,I5)
      WRITE(6,600)(TITLE(I1),I1=1,10)
 1400 CONTINUE
 2000 CONTINUE
```

```
      IF(PRPM.GT.0.0.AND.SWMP.GT.0.0)GO TO 1130
      GO TO 100
 1150 IF(PRPM.LE.0.0)GO TO 100
 1130 READ(5,560)NBRAN,(ISWEL(I),I=1,NBRAN)
      READ(5,485)NSWP,(SWRPM(I),I=1,NSWP)
  485 FORMAT(I5,7F10.0)
      SWMP=0.0
      GO TO 1140
  450 STOP 1001
  460 FORMAT(10A8)
  470 FORMAT(2I5,2F10.0)
  480 FORMAT (8F10.0)
  490 FORMAT(2I5,7F10.0)
  495 FORMAT(8F10.0)
  496 FORMAT(10X,70H INPUT DATA EXCEEDS ARRAY DIMENSIONS - PROGRAM TERMI
     1NATED                          ).
  500 FORMAT (1H1,44X,43HHYDRAULIC SYSTEM FREQUENCY RESPONSE PROGRAM,
     +//,30X,10A8)
```

APPENDIX C

TYPE 57 EMPIRICAL PUMP MODEL

HYTRAN USER MANUAL (AFAPL-TR-76-43, VOL. I)

6.57 TYPE #57 - EMPIRICAL PUMP MODEL

The Type #57 pump model will simulate pump outlet pressure and flow of a variable displacement pressure compensated pump. The user can select either a first order or second order computer model. The steady state operating characteristics are required input data. Empirically derived data provide the other input parameters.

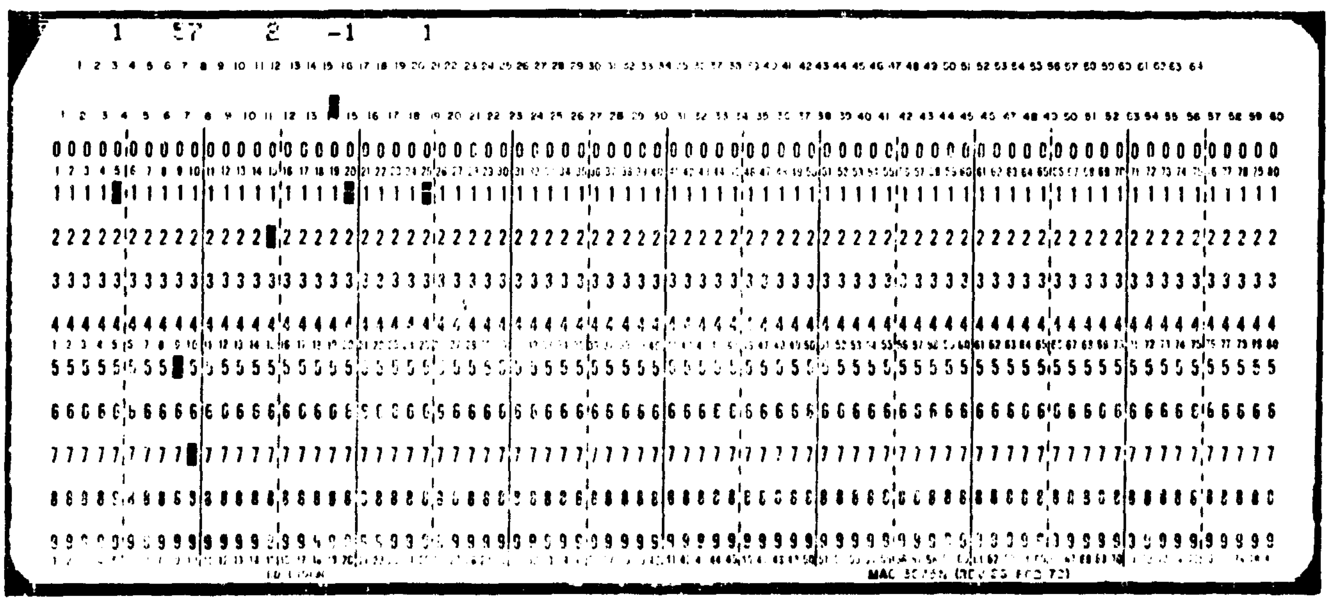
For the first order model a time constant dependent on the natural frequency of the torque response is input. Another time constant can be added to smooth the pressure response.

The second order model requires a natural frequency, damping factor, the final steady state flow, an approximate pump outlet volume (pumping chamber cavity) and a flow response delay. The input parameters can be measured or estimated from test data as discussed in AFWAL-TR-80-2039, (Note: The damping factor must be less than one.) The user can select a first or second order model by typing a zero or one in column 25 of card number 1 and entering the appropriate real data cards. If a second order model is chosen, a zero or one is placed in column 30 to denote either a turn-off or turn-on transient.

CARD NUMBER 1

COLUMN	FORMAT	DATA
1-5	I5	Component Number
6-10	I5	Type Number = 57
11-15	I5	Number of Real Data Cards = 1 or 2
16-20	I5	Line Number (with sign) attached to Connection 1
21-25	I5	0 = First Order Model 1 = Second Order Model
26-30	I5	0 = Turn-off transient 1 = Turn-on Transient
31-35	I5	
36-40	I5	
41-45	I5	
46-50	I5	
51-55	I5	
56-60	I5	
61-65	I5	
66-70	I5	
71-75	I5	
76-80	I5	Temperature/Pressure Code (See Page 4.0-2)

EXAMPLE CARD



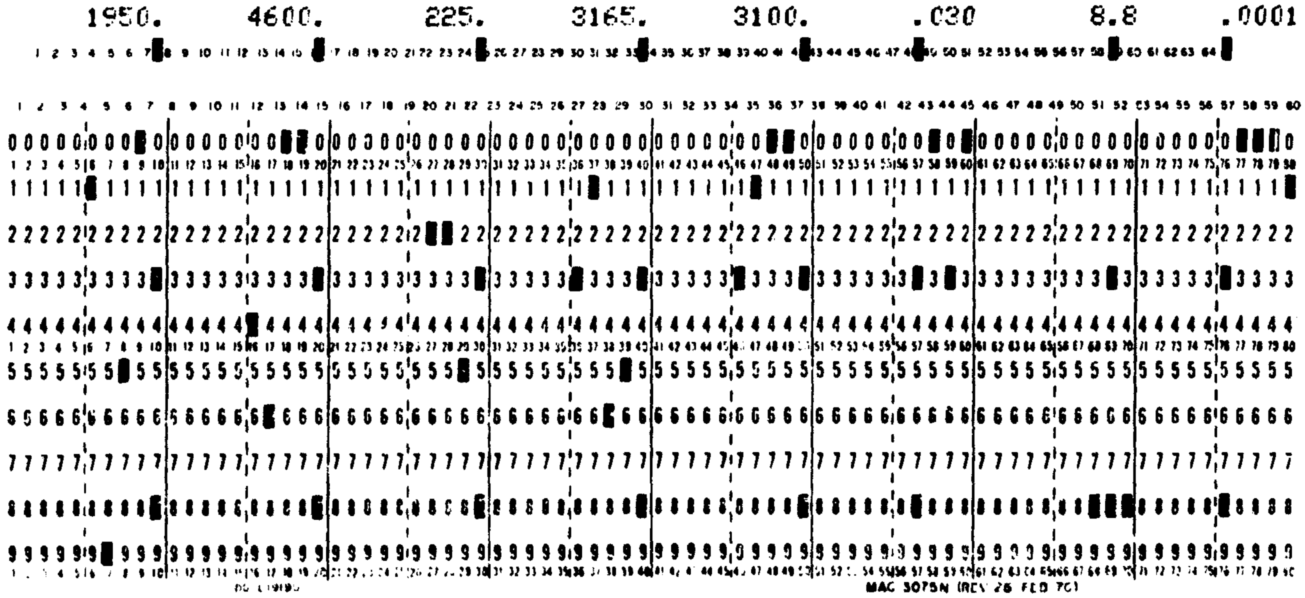
FIRST ORDER PUMP MODEL

CARD NUMBER 2

COLUMN	FORMAT	DATA	DIMENSIONS
1-10	E10.0	Pump Speed	RPM
11-20	E10.0	Pump Rated Speed	RPM
21-30	E10.0	Pump Flow @ Rated Speed	CIS
31-40	E10.0	Pressure @ Zero Flow	PSIA
41-50	E10.0	Pressure @ Full Flow	PSIA
51-60	E10.0	First Order Time Constant	SEC
61-70	E10.0	Case Drain Flow @ Rated Flow and Pressure	CIS
71-80	E10.0	Pressure Time Constant	SEC

EXAMPLE CARD

1950. 4600. 225. 3165. 3100. .030 8.8 .0001



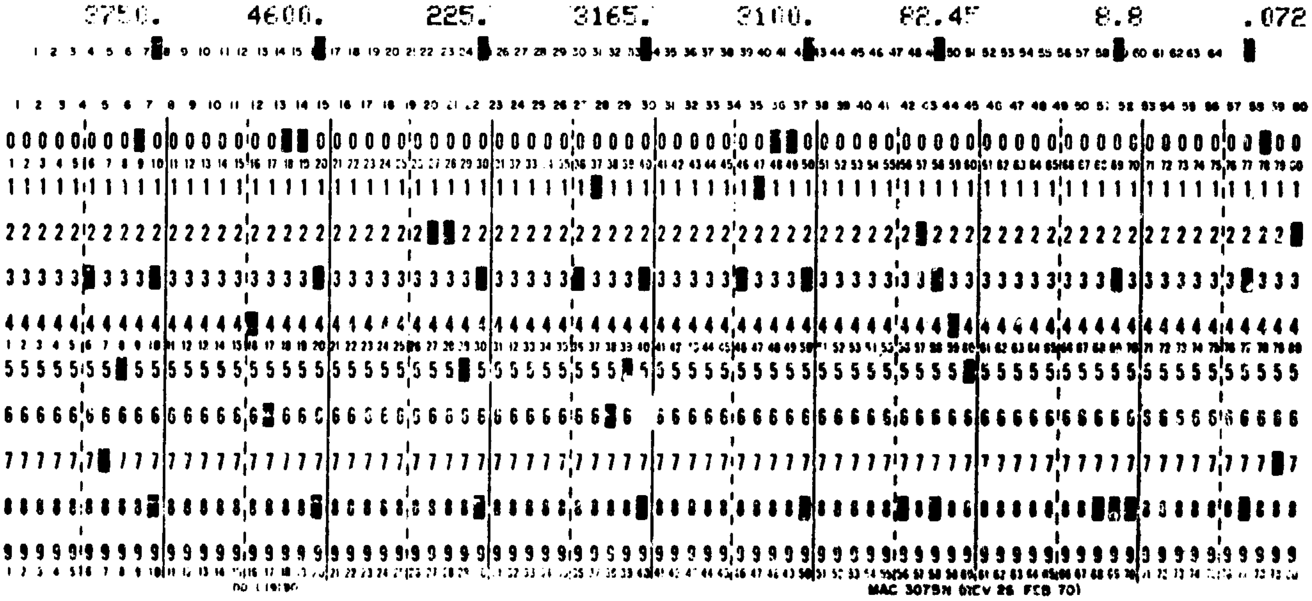
SECOND ORDER PUMP MODEL

CARD NUMBER 2

COLUMN	FORMAT	DATA	DIMENSIONS
1-10	E10.0	Pump Speed	RPM
11-20	E10.0	Pump Rated Speed	RPM
21-30	E10.0	Pump Flow @ Rated Speed	CIS
31-40	E10.0	Pressure @ Zero Flow	PSIA
41-50	E10.0	Pressure @ Full Flow	PSIA
51-60	E10.0	Natural Frequency	RAD/SEC
61-70	E10.0	Case Drain Flow @ Rated Flow and Pressure	CIS
71-80	E10.0	Damping Factor	-

EXAMPLE CARD

3750. 4600. 225. 3165. 3100. 82.45 8.8 .072

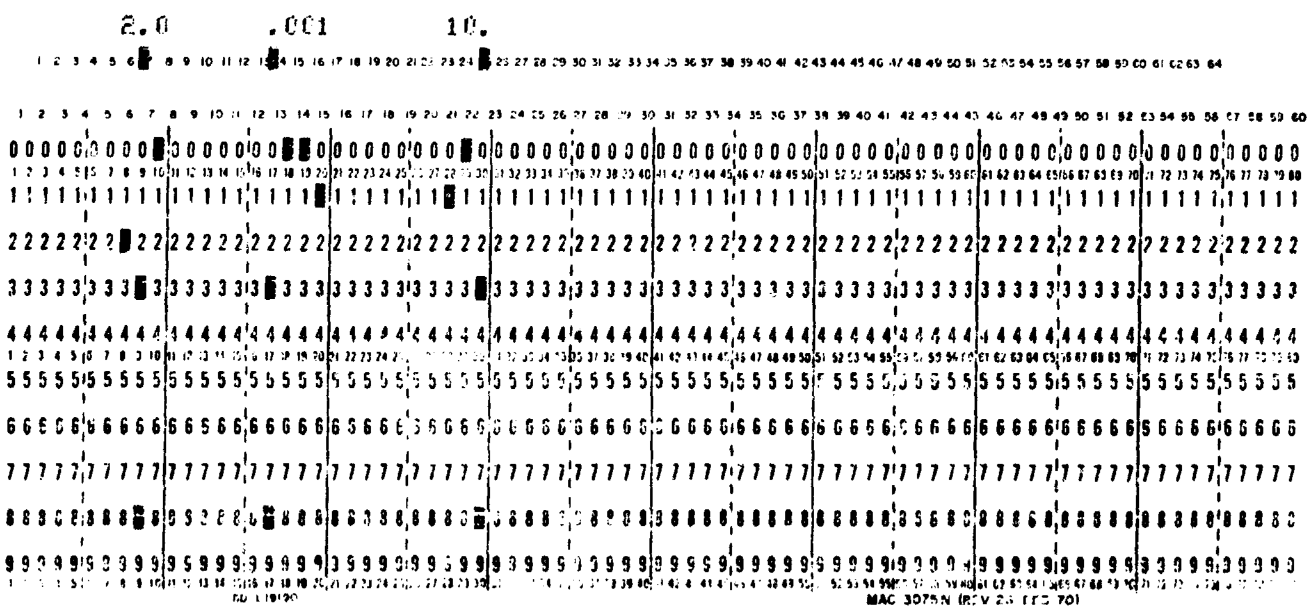


SECOND ORDER PUMP MODEL

CARD NUMBER 3

COLUMN	FORMAT	DATA	DIMENSIONS
1-10	E10.0	Final Steady State Flow	CIS
11-20	E10.0	Flow Response Delay	SEC
21-30	E10.0	Pump Outlet Volume	IN^3
31-40	E10.0		
41-50	E10.0		
51-60	E10.0		
61-70	E10.0		
71-80	E10.0		

EXAMPLE CARD



6.57 SUBROUTINE PUMP57

Subroutine PUMP57 models a variable displacement pressure compensated pump using empirically derived input data. The model is intended for use when detailed pump parameters for the more complex pump models are not known. The model will compute pump outlet pressure and flow and a torque value which reflects 100% pumping efficiency.

The PUMP57 subroutine contains both a first and second order pump model. Each model is selectable and requires the user to input the pump steady state operating characteristics. A time constant measured from test data is input for the first order model. The second order model requires an undamped natural frequency and damping factor, the steady state pump outlet flow after the transient, an approximate pump outlet volume, and a flow response delay.

The HYTRAN empirical pump model input parameters can be varied to achieve any degree of accuracy dependent upon the particular user application. However, when extrapolating the measured input data to other systems, caution should be exercised because of the sensitivity and interrelationship of the parameters. Actual data should be used whenever possible to verify the computed results. Knowing the closing or opening time of the transient control valve is also critical to the accuracy of the pump/system simulation.

6.57.1 MATH MODEL

In deriving the simplified models inherent physical properties of the pump are ignored. Non-linearities and distributed parameter which are present in an actual pump are estimated by linear lumped parameter models which result in ordinary differential equations with constant coefficients.

FIRST ORDER MODEL

A first order differential equation (1) can be written to define pump/system performance from the equivalent electrical schematic in Figure 6.57-1. The pump flow is represented by an ideal current source. The pump/system impedance is represented by a resistance and capacitance term.

$$Q_{IN} = \frac{P_{OUT}}{R_T} + C \frac{d\,P_{OUT}}{dt} \tag{1}$$

where $R_T = \dfrac{R_P R_L}{R_P + R_L}$

R_P = pump internal impedance $\dfrac{(PSI)}{(CIS)}$

R_L = load impedance $\dfrac{(PSI)}{(CIS)}$

C = pump/system capacitance $\dfrac{(IN^2 * IN)}{(PSI)}$

P_{OUT} = pump outlet pressure (PSI)

Q_{IN} = pump outlet flow (CIS)

Taking the Laplace transform of equation (1) and assuming zero initial conditions, the pump/system transfer function becomes

$$\frac{P_{OUT}(s)}{Q_{IN}(s)} = \frac{R_T}{R_T C s + 1} \tag{2}$$

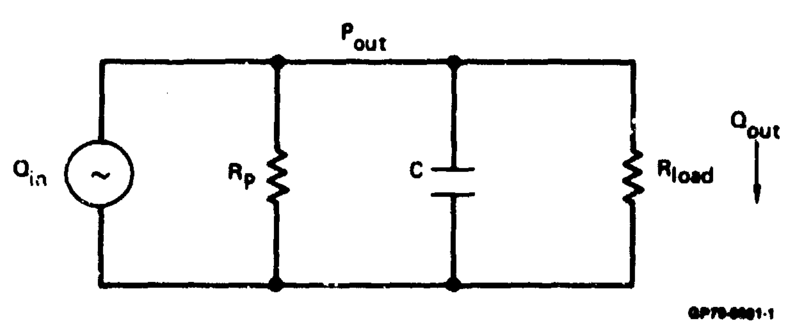


FIGURE 6.57-1 FIRST ORDER PUMP/SYSTEM MODEL EQUIVALENT SCHEMATIC

The load flow can be defined as

$$Q_{OUT}(s) = \frac{P_{OUT}(s)}{R_T\ (s)} \tag{3}$$

Solving equation (3) for $P_{OUT}(s)$ and substituting in equation (2), the pump/system transfer function is

$$\frac{Q_{OUT}(s)}{Q_{IN}\ (2)} = \frac{1}{Ts + 1} \tag{4}$$

where

T = first order time constant measured from test data (sec)

The first order pump model is represented by the diagram in Figure 6.57-2.

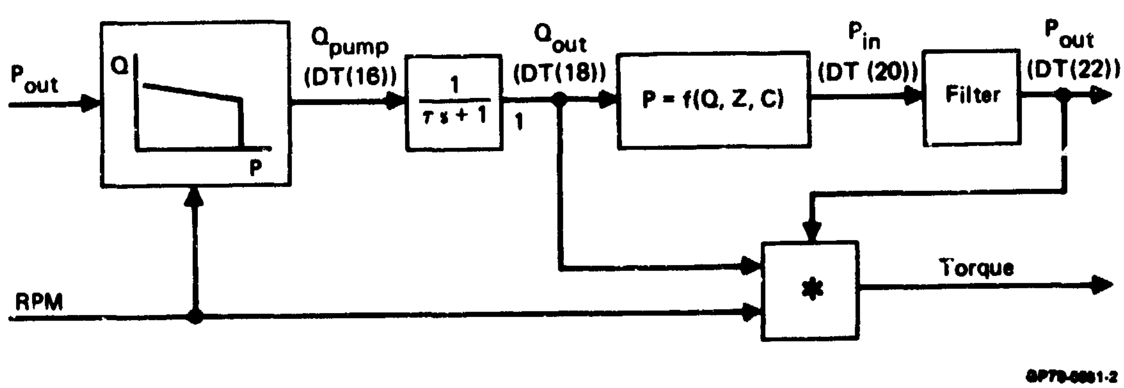


FIGURE 6.57-2 FIRST ORDER PUMP MODEL BLOCK DIAGRAM

System line pressure is used to determine pump outlet flow from the steady state pressure flow characteristics. When the pressure is less than full flow pressure, the pump outlet flow is

DT(16) = (DT(QT)-DT(RRQ))*(DT(POUT)-D(PFF))/(-D(PFF))+DT(RRQ) (5)

For system pressure greater than or equal to full flow pressure, the outlet flow is

DT(16) = DT(RRQ)*(DT(POUT)-D(POF))/(D(PFF)-D(POF)) (6)

The flow (DT(16)) then passes through the first order lag network.

DT(18) = DYNAM(DT(15),DT(17),DT(5)) (7)

The resulting pump flow (DT(18)) is combined with the line dynamics to compute a pump outlet pressure.

DT(20) = C(L1) + DT(18) * Z(L1) (8)

The pressure is passed through a first order lag circuit usually with a small time constant to eliminate any high frequency pressure spikes.

DT(22) = DYNAM(DT(19), DT(21), DT(12)) (9)

The pump operating RPM is combined with the outlet pressure and flow to obtain a torque.

DT(ATORQ) = DT(POUT) * DT(QOUT) *D(RPM) (10)

SECOND ORDER MODEL

The second order model more closely emulates the mechanical action of the pump. Variable displacement pumps usually incorporate a control valve which strokes a piston that moves the pump hanger changing the output flow rate. All these functions including the feedback pressure can be broken down into individual functions, then pieced together to form a model.

The pump compensator valve will initially sense a change in system pressure. A definite lag time is involved until the compensator starts to move and ports fluid to or from the hanger actuator. The response of the control spool is important for it controls the rate at which the hanger will respond and eventually dampen to a final flow value. The delay time can be easily measured as the time the output pressure changes to the time that the torque changes.

The control spool position and the hanger actuator position can be related through a second order response. The actuator position is integrated to obtain a velocity which moves the hanger to a pump flow. The flow reacts with the system pressure providing a feedback through the compensator network.

In the pump model the second order response was used to simulate the control spool and hanger dynamics. The transfer function output was multiplied by a proportional controller to obtain the outlet flow. Feedback to the second order transfer function was provided by outlet pressure referenced to the final steady state pressure value.

Damping and frequency values for the second order transfer function can be obtained from the transient torque characteristics. Steady state pump pressure/flow characteristics are used to determine the pump model final steady state values.

A schematic diagram of the pump model is found in Figure 6.57-3. The model has no pressure feedback loop to the control valve. Once the transient is started the second order function will automatically propagate at the selected natural frequency and damping rate.

DT(RESPON) = SECORD(1,XSIG,D(OMEGA),D(ZETA),DELT,DT(ARR)) (11)

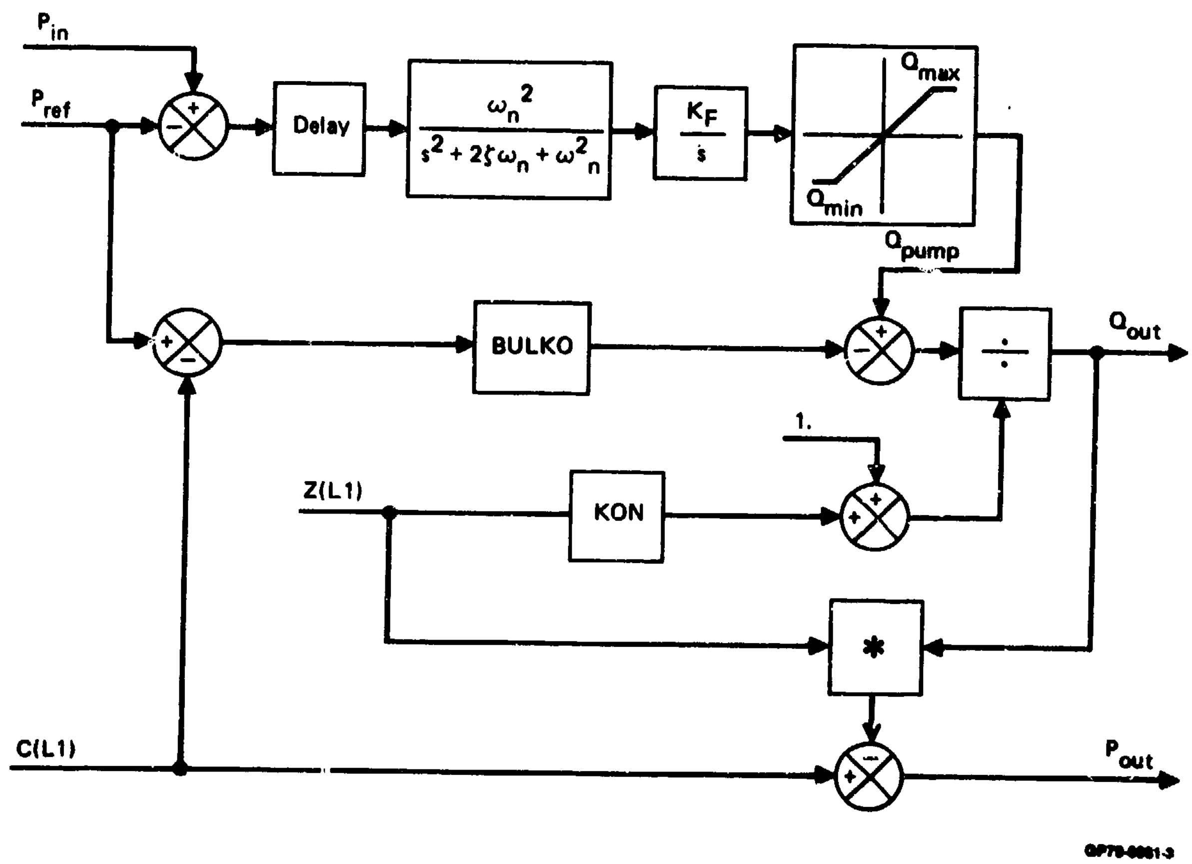


FIGURE 6.!7-3 SECOND ORDER PUMP MODEL BLOCK DIAGRAM

A pressure reference is provided by the compressibility flow (Q_{comp}). The flow is combined with the pump outlet flow as a correction factor.

DT(QOUT) = (DT(FLOW)+DT(NEWP)*DT(BULKO)-C(L1)*DT(KON))/(1.+Z(L1)*DT(KON)) (12)

The resultant flow is then solved with the line boundary equation to obtain a pump outlet pressure.

DT(POUT) = C(L1) + DT(QOUT) * Z(L1) (13)

6.57.2 ASSUMPTIONS

In the derivation of the second order response it is assumed that the transient control valve operation can be approximated by a step input, and the valve completely closes off the line. This is usually not the case in practical hydraulic design. Oil leaks past the valve and energy dissipation rises. The net result are changes in the effective natural frequency and damping of the system. The effect on frequency is considerable but that on damping can be by a factor of ten. Calculations of system performance for one operating condition in a system may be in error for other and not very different conditions. This is due mainly to the non-linearity of fluid power systems and the care in selecting system variables cannot be overemphasized.

6.57.3 COMPUTATIONS

The pump subroutines is set up using the HYTRAN program commons plus the D, DT, DD and L arrays.

1000 SECTION

In the 1000 section the rated flow at the operating RPM and the total pump flow are computed. The first or second order functions are initialized depending on which model the user selected.

1500 SECTION

In this section the pump outlet pressure is calculated for the steady state portion of the HYTRAN program. The outlet pressure is computed using the input steady state pump characteristics and the current flow guess.

2000 SECTION

With the completion of the steady state calculations, all the variables for use in the transient portion of the program are initialized.

3000 SECTION

In the 3000 section the transient response of the pump is computed.

A check is made to determine whether a first or second order model was selected. For the first order model the pump outlet flow is computed and passed through a first order lag with a call to the function DYNAM. The subsequent flow is then used to calculate the pump outlet pressure and pump torque.

When the second order model is selected, the response function SECORD, is not called until the rise or fall in system pressure exceeds a deadband value.

```
IF(ABS(DT(DELTAP)).LT.DT(DEADBD))GO TO 3200
```
(14)

The deadband pressure is the difference between the pumps zero flow and full flow pressures.

Once the transient begins, a counter (L(4)) is set, and the delay time (D(DELAY)) is added to the current HYTRAN program time. After the delay times out, the transient response begins. The total pump outlet flow is based on the response function SECORD.

Since the forcing function is a unit step, the output of SECORD will start at zero and end at one, with excursion above one dependent on the damping factor and natural frequency selected by the user. The outlet flow is a percentage of a value between the user input maximum and minimum flow values or

```
DT(FLOW) = (DT(QMAX)-DT(QMIN))*(DT(RESPON))+DT(QMIN)
```
(15)

For a turn-off transient the response is

```
DT(RESPON) = 1. - DT(RESPON)
```
(16)

and for a turn-on transient the response is the functional value.

The pump outlet flow and pressure are computed as described in the math model section (Section 6.57.1).

6.51.4 APPROXIMATION

Both the first and second order models are attempts to describe a complex undamped servo system. In the simplified models effects were included which would yield reasonable results when compared to experimental data. At best they are approximations to the actual pump operation and in many respects do not resemble the internal workings of the pump at all.

6.51.5 LIMITATIONS

The models were derived to provide quick,ballpark answers without the benefit of detailed design data. The models should not be used when detailed pump/system performance prediction is critical.

The first order model is inadequate in small volume systems with a fast closing valve (turn-off transient). A large time constant for the pressure filter is required to dampen the resultant pump pressure output. A second order model will provide a more stable response.

The computer program input data is dependent on measured test data. However, once the input parameters have been established, they can be applied to other system simulations.

Both empirical models simulate outlet pressure and flow response to turn-on and turn-off transients.

6.57.6 VARIABLE NAMES

NAME	DESCRIPTION	DIMENSION
DT(ARR)	Dummy Variable	-
DT(ATORQ)	Pump Torque	IN-LB
DT(BEGPR)	Pump Steady State Outlet Pressure	PSI
DT(BULKO)	Compressibility Constant	CIS/PSI
DT(DEADBD)	Zero Flow Outlet Pressure Minus Full Flow Outlet Pressure	PSI
D(DELAY)	Second Order Time Delay	SEC
DT(DELTAP)	Outlet Pressure Difference from Steady State Value	PSI

NAME	DESCRIPTION	DIMENSION
DT(FLOW)	Total Pump Outlet Flow	CIS
DT(KON)	Dummy Variable	-
DT(LEAK)	Dummy Variable	-
DT(NEWP)	Dummy Variable	-
D(OMEGA)	Natural Frequency	RAD/SEC
D(POF)	Outlet Pressure at Zero Flow	PSI
DT(POUT)	Outlet Pressure	PSI
DT(PPOUT)	Previous Outlet Pressure	PSI
DT(PRESS)	Final Steady State Pump Outlet Pressure	PSI
D(PVOL)	Pump Outlet Volume	IN^3
QA	Steady State Outlet Flow	CIS
D(QFINAL)	Final Steady State Flow	CIS
D(QLK)	Case Drain Flow	CIS
DT(QMAX)	Maximum Pump Flow	CIS
DT(QMIN)	Minimum Pump Flow	CIS
DT(QOUT)	Pump Outlet Flow	CIS
QS	Flow Sign	-
DT(QT)	Total Pump Flow	CIS
DT(RESPON)	Second Order Response	-
D(RPM)	Pump Operating Speed	RPM
D(RQ)	Rated Flow	CIS
D(RRPM)	Rated Operating Speed	RPM
DT(RRQ)	Rated Flow at the Operating Speed	CIS
D(TAU)	First Order Flow Time Constant	SEC
D(TAU1)	First Order Pressure Time Constant	SEC
DT(TIMD)	Dummy Variable	-
D(ZETA)	Damping Factor	-

6.57.7 SUBROUTINE LISTING

```
      SUBROUTINE PUMP57 (D DT,DD,L)
C     ***** REVISED OCTOBER 1979 *****
C     HYTRAN EMPIRICAL PUMP MODEL - PUMP57
C     FIRST AND SECOND ORDER      RESPONSE
C
      COMMON NTELPL,NTOLPL,IPT,IPOINT,NPTS,INEL,KNEL,NTOPL,NLPLT(61,3),
     1 PQLEG(90,12),LCS(90,10),ILEG(1400),PN(90),QN(90),PEX(90)
      COMMON/SUB/PARM(150,9),PM(1500),QM(1500),P(300),Q(300),C(300)
     1,Z(300),RHO(20),S2ORHO(20),VISC(20),BULK(20),TEMP(20),PVAP(20)
     2,ATPRES,T,DELT,TFINAL,PLTDEL,PI,TITLE(20),LEGN,ICON
     3,KTEMP(99),LSTART(150),NLPT(150),LTYPE(99),NC(99),INX,INZ
     4,INV,ISTEP,NLINE,NEL,IND,IENTR,MNLINE,MNEL,MNLEG,MNNODE,MNPLOT
     5,MNLPTS,MDS,YSCAL(61,2)
      DIMENSION D(20),DT(30),DD(1),L(5)
      INTEGER RPM,RRPM,RQ,POF,PFF,QMAX,POUT,QMIN,TIMD,QOUT,TAU,TAU1
     + ,OMEGA,RRQ,QLK,QT,ZETA,ATORQ,DELTAP,DEADBD,PRESS,PVOL
     + ,QFINAL,DELAY,RESPON,PPOUT,FLOW,BULKO,ARR,NEWP,BEGPR
C     D( ) ARRAY *****
      DATA RPM/1/,RRPM/2/,RQ/3/,POF/4/,PFF/5/,OMEGA/6/,QLK/7/,ZETA/8/
     + ,QFINAL/9/,DELAY/10/,PVOL/11/,TAU/6/,TAU1/8/
C     DT( ) ARRAY *****
      DATA POUT/1/,QOUT/2/,DELTAP/3/,ATORQ/4/,RESPON/5/,TIMD/6/,
     + DEADBD/7/,QMIN/8/,QMAX/9/,RRQ/10/,QT/11/,PPOUT/12/
     + ,LEAK/13/,FLOW/14/,BULKO/15/,KON/16/,PRESS/17/
     + ,NEWP/18/,BEGPR/19/,ARR/20/
C
C       L(2)=0 FIRST ORDER MODEL  :  L(2)=1 SECOND ORDER MODEL
C
C                                    L(3)=0 TURN-OFF TRANSIENT
C                                    L(3)=1 TURN-ON TRANSIENT
C
      IF(IENTR) 1000,2000,3000
C *** 1000 SECTION
 1000 CONTINUE
      IF(INEL.NE.0)GO TO 1500
      DO 1001 I=1,30
 1001 DT(I)=0.0
      L(4)=0
      DT(RRQ)=D(RQ)*D(RPM)/D(RRPM)
      DT(QT)=D(QLK)+DT(RRQ)
      IF(L(2).EQ.0)GO TO 1010
      DT(RESPON)=SECORD(0,0.,D(OMEGA),D(ZETA),DELT,DT(ARR))
      RETURN
 1010 CALL TUSTIN(2,D(TAU),DT(5),DELT)
      CALL TUSTIN(2,D(TAU1),DT(12),DELT)
      RETURN
C
C      STEADY STATE CALCULATION SECTION
C
```

6.57.7 (Continued)

```
 1500 CONTINUE
      QA=PQLEG(INEL,1)
      QS=PQLEG(INEL,2)
      IF(QA*QS.LT.0.0)QA=0.0
      IF(QA.GT.DT(QT))QA=DT(QT)
C     IF(QA.GT.DT(RRQ))GO TO 1600
      DT(POUT)=D(POF)-((D(POF)-D(PFF))*QA/DT(RRQ))
      GO TO 1700
 1600 DT(POUT)=D(PFF)-(D(PFF)*(QA-DT(RRQ))/(DT(QT)-DT(RRQ)))
 1700 CONTINUE
      N=LCS(INEL,2)
      QN(N)=DT(POUT)*1000.-QA*QS
      PEX(N)=1000.
      LCS(INEL,7)=5
      RETURN
C *** 2000 SECTION
 2000 CONTINUE
      D(RPM)=30./(D(RPM)*PI)
      IF(L(2).EQ.0)GO TO 2200
      DT(BEGPR)=DT(POUT)
      DT(NEWP)=DT(POUT)
      DT(PPOUT)=DT(POUT)
      DT(QOUT)=-Q(L(1))
      DT(FLOW)=DT(QOUT)
      DT(BULKO)=D(PVOL)/(BULK(KTEMP(IND))*DELT)
      DT(KON)=DT(BULKO)
      DT(ATORQ)=DT(POUT)*DT(QOUT)*D(RPM)
      DT(DEADBD)=D(POF)-D(PFF)
      DT(QMIN)=D(QFINAL)
      DT(QMAX)=DT(QOUT)
      IF(L(3).EQ.0)GO TO 2100
      DT(QMIN)=DT(QOUT)
      DT(QMAX)=D(QFINAL)
 2100 CONTINUE
      DT(PRESS)=D(POF)-((D(POF)-D(PFF))*D(QFINAL)/DT(RRQ))
      IF(DT(PRESS).GT.D(POF))DT(PRESS)=D(POF)
      GO TO 2300
 2200 CONTINUE
      DO 2210 I=15,18
      DT(I)=-Q(L(1))
 2210 DT(I+4)=DT(POUT)
      DT(ATORQ)=DT(POUT)*DT(15)*D(RPM)
 2300 CONTINUE
C     WRITE(6,900)(DT(I),I=1,30)
  900 FORMAT(1X,10E12.5)
      RETURN
```

6.57.7 (Continued)

```
 3000 CONTINUE
      L1=L(1)
      IF(L(2).EQ.0)GO TO 3400
      DT(POUT)=C(L1)-Q(L1)*Z(L1)
      IF(L(4).EQ.1) GO TO 3100
      DT(DELTAP)=DT(POUT)-DT(BEGPR)
      IF(ABS(DT(DELTAP)).LT.DT(DEADBD))GO TO 3200
      L(4)=1
      DT(TIMD)=T+D(DELAY)
 3100 CONTINUE
C       TURN-ON TRANSIENT
      XSIG=1.
      IF(T.LE.DT(TIMD))GO TO 3300
      DT(RESPON)=SECORD(1,XSIG,D(OMEGA),D(ZETA),DELT,DT(ARR))
C       TURN-OFF TRANSIENT
      IF(L(3).EQ.0)DT(RESPON)=1.-DT(RESPON)
      DT(FLOW)=(DT(QMAX)-DT(QMIN))*(DT(RESPON))+DT(QMIN)
 3200 CONTINUE
      DT(NEWP)=DT(BEGPR)
      IF(L(4).EQ.1)DT(NEWP)=DT(PRESS)
 3300 CONTINUE
      DT(QOUT)=DT(FLOW)+DT(NEWP)*DT(BULKO)-C(L1)*DT(KON)
      DT(QOUT)=DT(QOUT)/(1.+Z(L1)*DT(KON))
      DT(POUT)=C(L1)+DT(QOUT)*Z(L1)
      DT(PPOUT)=DT(POUT)
      GO TO 3500
 3400 CONTINUE
      IF(DT(POUT).LT.D(PFF))GO TO 3420
      DT(16)=DT(RRQ)*(DT(POUT)-D(POF))/(D(PFF)-D(POF))
      GO TO 3440
 3420 CONTINUE
      DT(16)=(DT(QT)-DT(RRQ))*(DT(POUT)-D(PFF))/(-D(PFF))+DT(RRQ)
 3440 CONTINUE
      DT(18)=DYNAM(DT(15),DT(17),DT(5))
      DT(20)=C(L1)+DT(18)*Z(L1)
      DT(22)=DYNAM(DT(19),DT(21),DT(12))
      DT(19)=DT(20)
      DT(21)=DT(22)
      DT(17)=DT(18)
      DT(15)=DT(16)
      DT(POUT)=DT(22)
      DT(QOUT)=DT(18)
 3500 CONTINUE
      P(L1)=DT(POUT)
      Q(L1)=-DT(QOUT)
      DT(ATORQ)=DT(POUT)*DT(QOUT)*D(RPM)
      RETURN
      END
```

6.51 TYPE #53 - F-15 PUMP

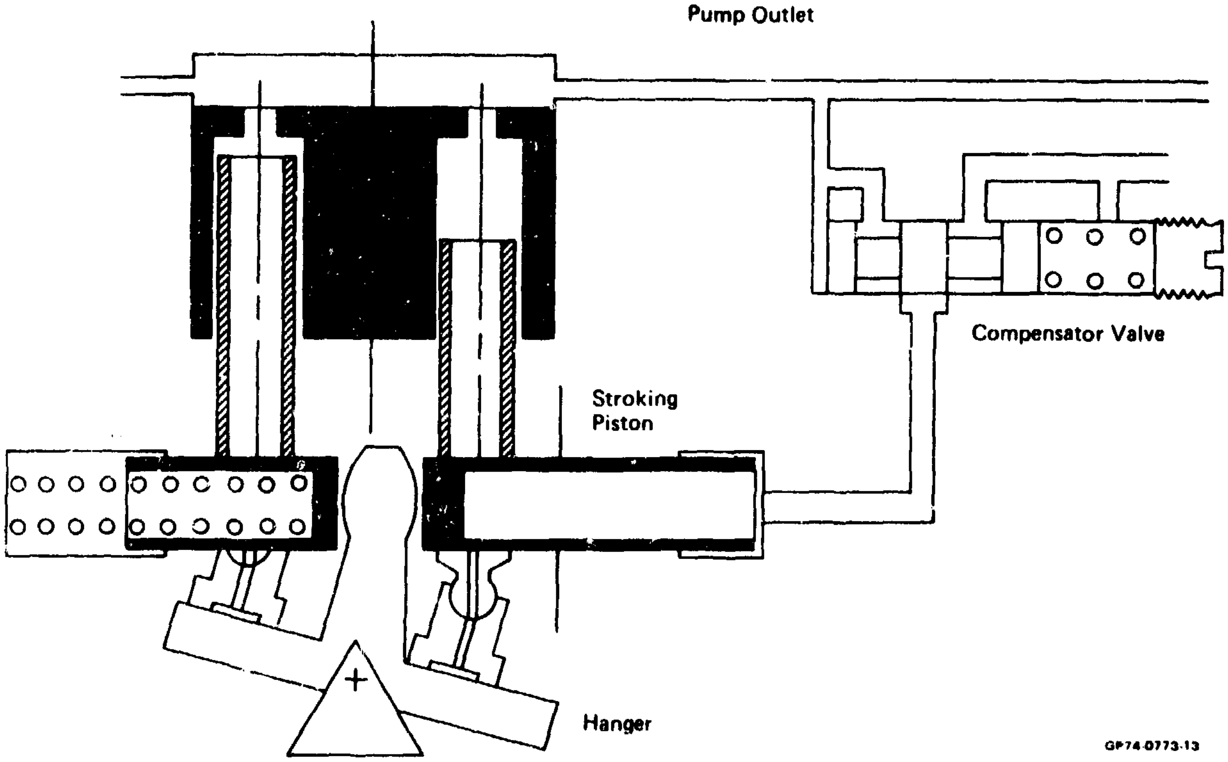


FIGURE 6.53-1
TYPE NO. 53 PRESSURE REGULATED VARIABLE DISPLACEMENT PUMP

The Type 53 model which simulates an Abex F-15 hydraulic pump is a complicated subroutine. The F-15 pump dynamic characteristics are sufficiently complex to warrant the special treatment.

The PUMP53 model can be used to model other pressure compensated variable displacement pumps. When modifying the pump variables, the user should be careful. A pump is essentially a complex underdamped servo system which is prone to instability, and it is easy to make it worse.

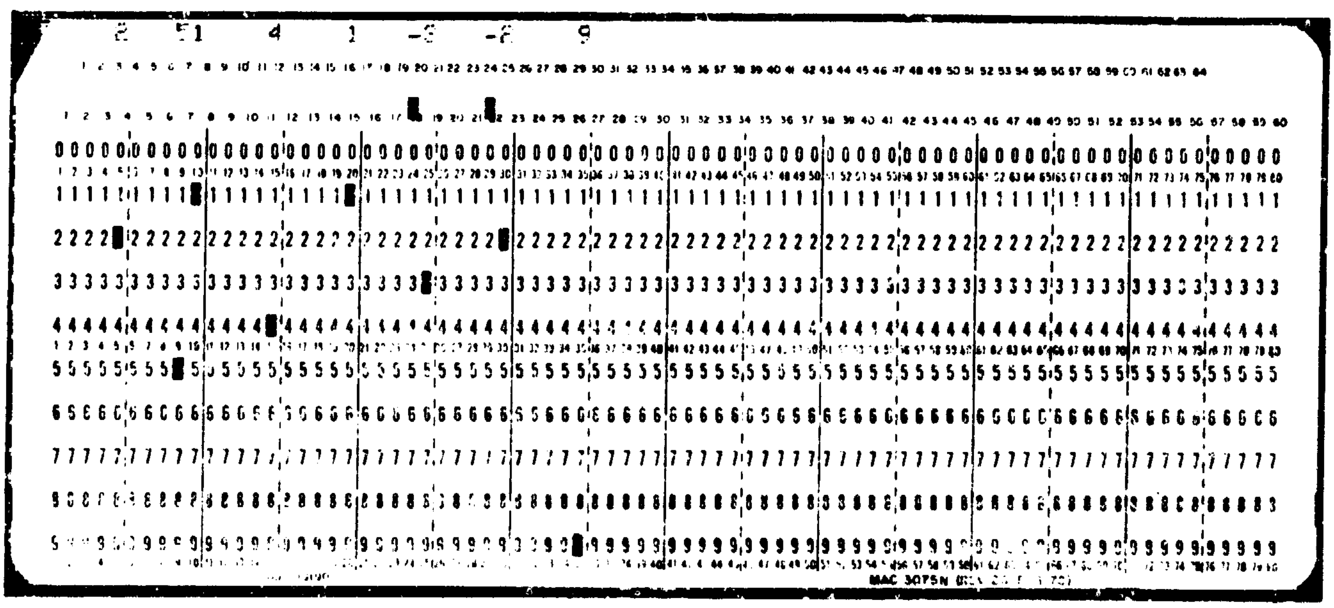
The ABEX F-15 pump has a fast response going from 10% to 90% stroke in approximately 15 milliseconds, so the user should take care in designing the system, to avoid cavitation problems, caused by rapidly changing flow demands in the suction lines.

The F-15 pump input data is specific to that pump and cannot be used for other pumps.

CARD NUMBER 1

COLUMN	FORMAT	DATA
1-5	I5	Component Number
6-10	I5	Type Number = 53
11-15	I5	Number of Real Data Cards = 4
16-20	I5	Line Number (with sign) attached to Connection 1 (Inlet)
21-25	I5	Line Number (with sign) attached to Connection 2 (Outlet)
26-30	I5	Line Number (with sign) attached to Connection 3 (Case Drain)
31-35	I5	Number of Pistons
36-40	I5	
41-45	I5	
46-50	I5	
51-55	I5	
56-60	I5	
61-65	I5	
66-70	I5	
71-75	I5	
76-80	I5	Temperature/Pressure Code (See Page 4.0-2)

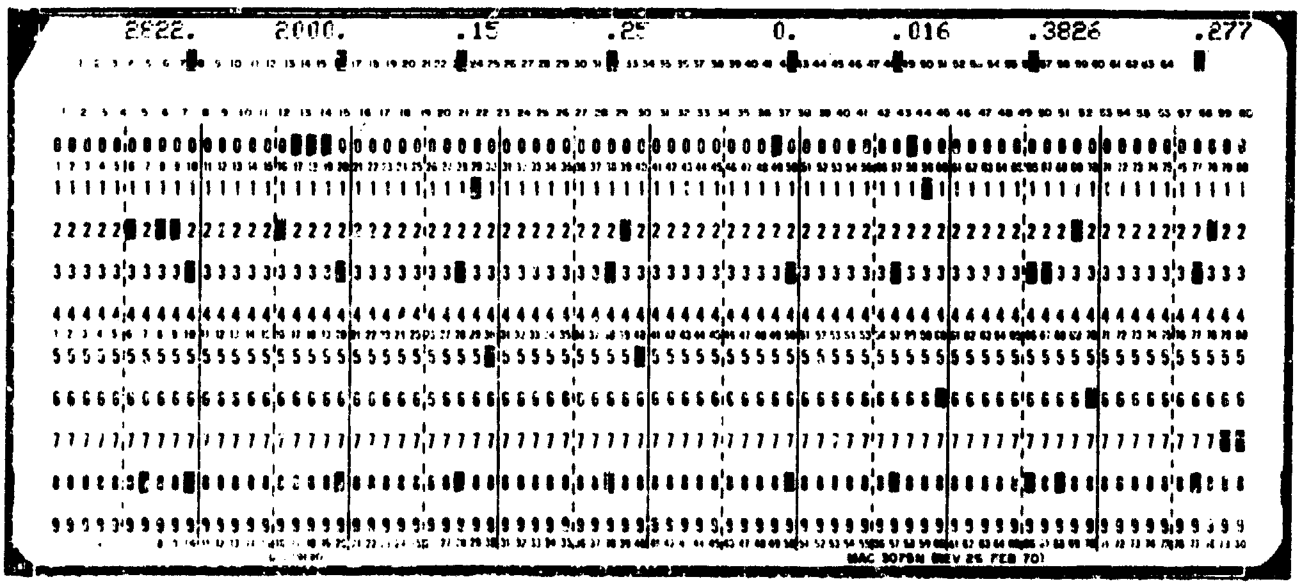
EXAMPLE CARD



CARD NUMBER 2

COLUMN	FORMAT	DATA	DIMENSIONS
1-10	E10.0	Pressure at which valve begins to open from actuator to case	PSIA
11-20	E10.0	Valve spring rate	LBS/IN
21-30	E10.0	Compensator Valve Area	IN**2
31-40	E10.0	Slot width	IN
41-50	E10.0	Blank	-
51-60	E10.0	Valve overlap	IN
61-70	E10.0	Piston Area	IN**2
71-80	E10.0	Hanger Actuator Lever Arm from Hanger Pivot at Mid Stroke	IN

EXAMPLE CARD



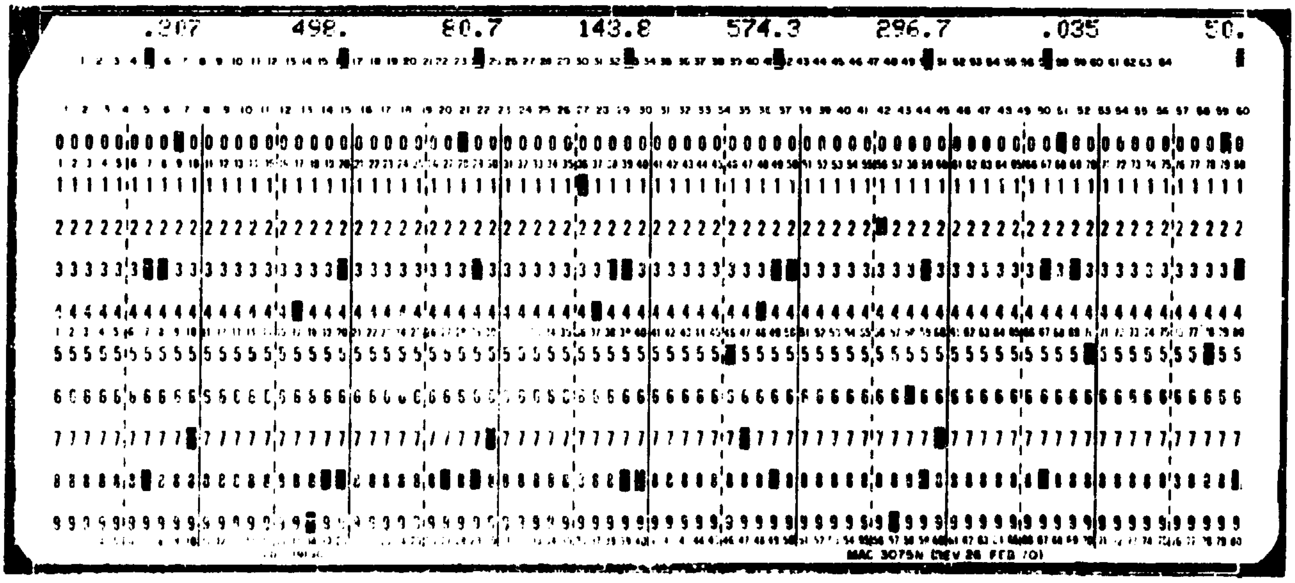
CARD NUMBER 3

COLUMN	FORMAT	DATA	DIMENSIONS
1-10	E10.0	Actuator Area	IN**2
11-20	E10.0	Actuator Pressure Due to Spring Force at Zero Pump Displacement	PSI
21-30	E10.0	Actuator Pressure due to Spring Force at Maximum Pump Displacement	PSI
31-40	E10.0	Actuator Pressure Due to Piston Acceleration @ 3600 RPM	IN**2/SEC
41-50	E10.0	Actuator Pressure Inputed at 3600 RPM and Zero Pump Displacement +	PSI
51-60	E10.0	Actuator Pressure at 3600 RPM and Maximum Pump Displacement +	PSI
61-70	E10.0	Slope of Pressure vs RPM Curve +	PSI/RPM
71-80	E10.0	Hanger Damping*	PSI/IN/SEC

* = Referenced to Actuator Pressure

\+ = Excluding pressure due to the spring and pumping piston acceleration.

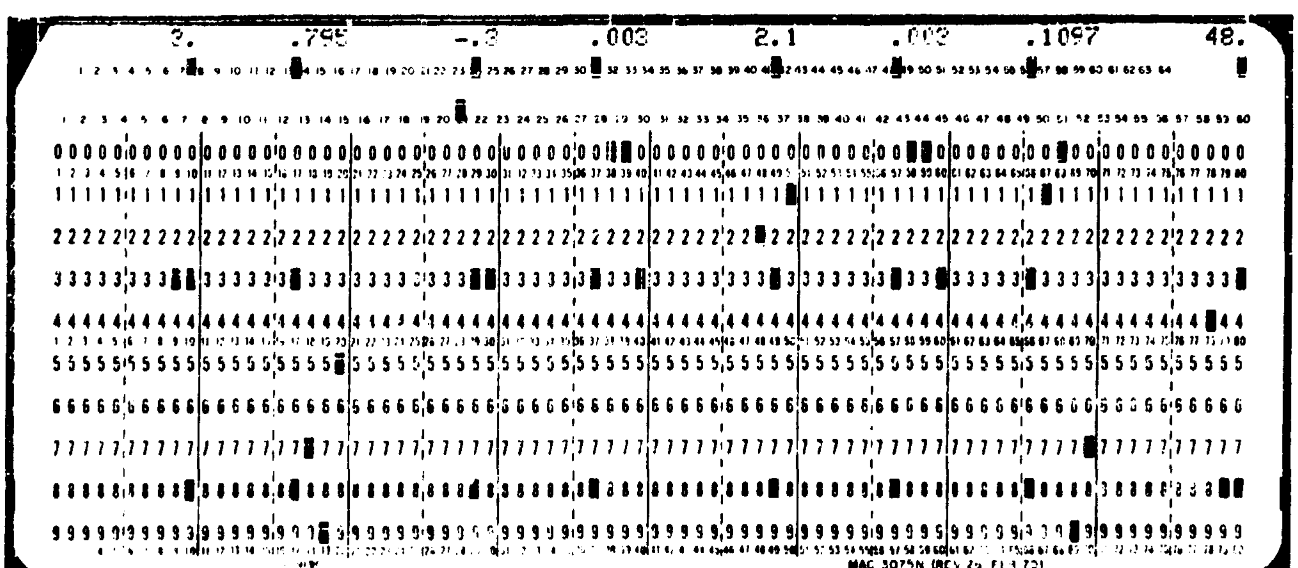
EXAMPLE CARD



CARD NUMBER 4

COLUMN	FORMAT	DATA	DIMENSIONS
1-10	E10.0	Theoretical Maximum Pump Displacement	IN**3/REV
11-20	E10.0	Maximum Actuator Displacement @ Maximum Flow	IN
21-30	E10.0	Minimum Actuator Displacement @ Minimum Pump Flow (-ve)	IN
31-40	E10.0	Flat Depth	IN
41-50	E10.0	Minimum Actuator Engagement	IN
51-60	E10.0	Coefficient of Pump Leakage	CIS/PSI
61-70	E10.0	Coefficient of Leakage from Case to Inlet	CIS/PSI
71-80	E10.0	Case Volume	IN**3

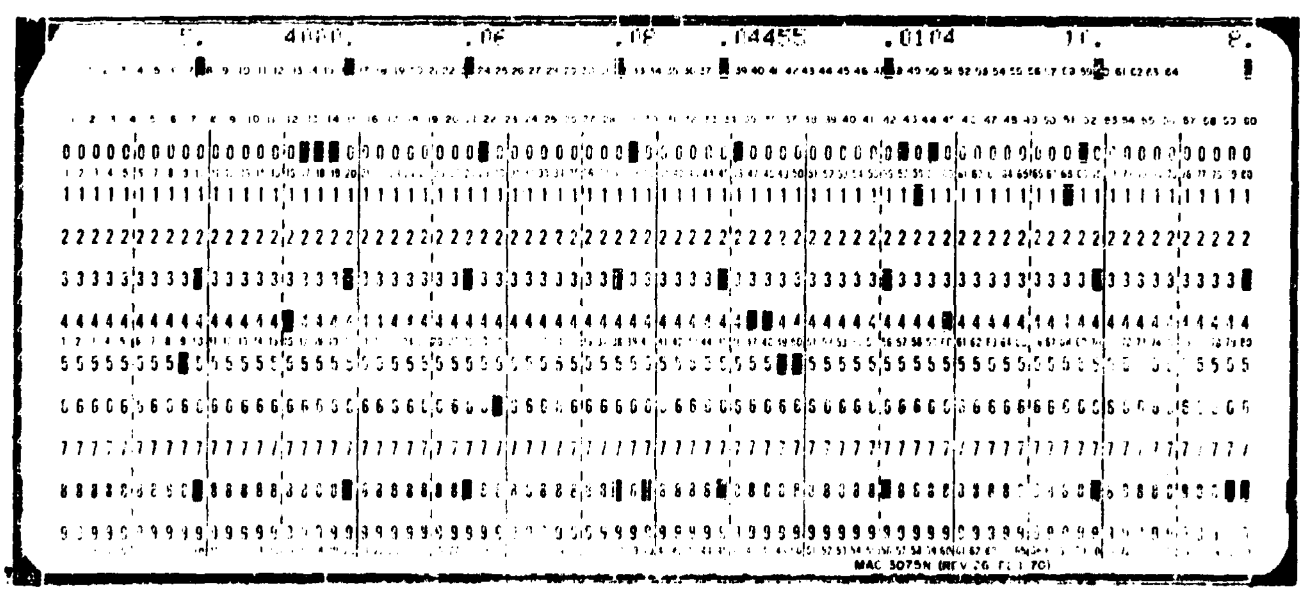
EXAMPLE CARD



CARD NUMBER 5

COLUMN	FORMAT	DATA	DIMENSIONS
1-10	E10.0	Minimum Inlet Pressure	PSI
11-20	E10.0	Pump Operating Speed	RPM
21-30	E10.0	Hanger Offset	IN
31-40	E10.0	Maximum Compensator Valve Displacement	IN
41-50	E10.0	Case Drain Port Area	IN**2
51-60	E10.0	Rotating Group Mass	LBS-SEC**2/IN
61-70	E10.0	Actuator Volume	IN**3
71-80	E10.0	Outlet Volume	IN**3

EXAMPLE CARD



6.53 SUBROUTINE PUMP53

Subroutine PUMP53 was set up to model an F15 pump, which is basically a simple in line piston pump, though it incorporates many mechanical refinements and sophisticated design features. The model is intended for use by system designers and is primarily aimed at the study of pump system stability under dynamic loading conditions.

The model incorporates the effects of case drain dynamics, since the F15 pump output pressure is referenced to case and the actuator discharges to the case, and displaces case volume when it is moving. The treatment of leakage and damping characteristics are rudimentary.

The dynamics of the hanger are complex. For the model the effects of the dynamic forces on the actuator are included. These forces push the pump to maximum flow, the hanger spring provides this force on start-up.

In addition the hanger offset creates a negative flow at the pump inlet and outlet when the hanger is moving toward maximum flow, and this has a destabilizing effect, when the hanger response is very fast.

Some of the hanger forces are oscillatory but no attempt has been made to describe this effect, except that the magnitude is sufficient to keep the hanger in motion, so the effects of static friction is ignored. This is an assumption that helps the simulation by keeping the integration of the hanger velocity a continuous function, between its mechanical stops.

The compensator valve dynamics are a significant part of the model. The forces on the valve are a combination of the outlet pressure force pushing against the case pressure and spring forces, with damping and flow forces acting in either direction.

The damping and flow forces are estimates using classical formulae.

Under certain conditions, the pump compensator valve responds to the oscillatory pressures caused by the pump pulsations. While no attempt has been made to describe how these are generated, provisions have been added so their effect on the compensator can be determined.

The compensator valve on the F15 is overlapped, but under certain conditions the pulsations can cause the valve to dither, reducing the effect of the overlap and changing the response characteristics of the pump.

The damping and flow forces are included because of the fast response of the compensator valve.

6.53-1 <u>Math Model</u>

A simplified diagram of the pressure regulated variable displacement pump is shown in Figure 6.53-1.

An equivalent circuit schematic diagram for the pump model is shown in Figure 6.53-2.

<u>Pump Displacement Flow</u>

For the pump inlet the displacement flow is computed as follows:

QPUMP = D(DISP)*DT(PRPM)*(DISACT)

if D(DISAMI)<DT(DISACT)<D(DISAM) WHERE D(DISP) = D(DISP)/60./D(DISAM)

or QPUMP = QINLET + QCASIN

if PINLET≤PINMIN

<u>Actuator Pressure</u>

The actuator pressure is based on the contributions of the spring force, case pressure, outlet pressure and pump rpm, plus the reaction force due to velocity damping which is generated when the hanger is moving. The input data establishing actuator pressure for pump operating conditions is modified to give a simpler algorithm for the transient calculations.

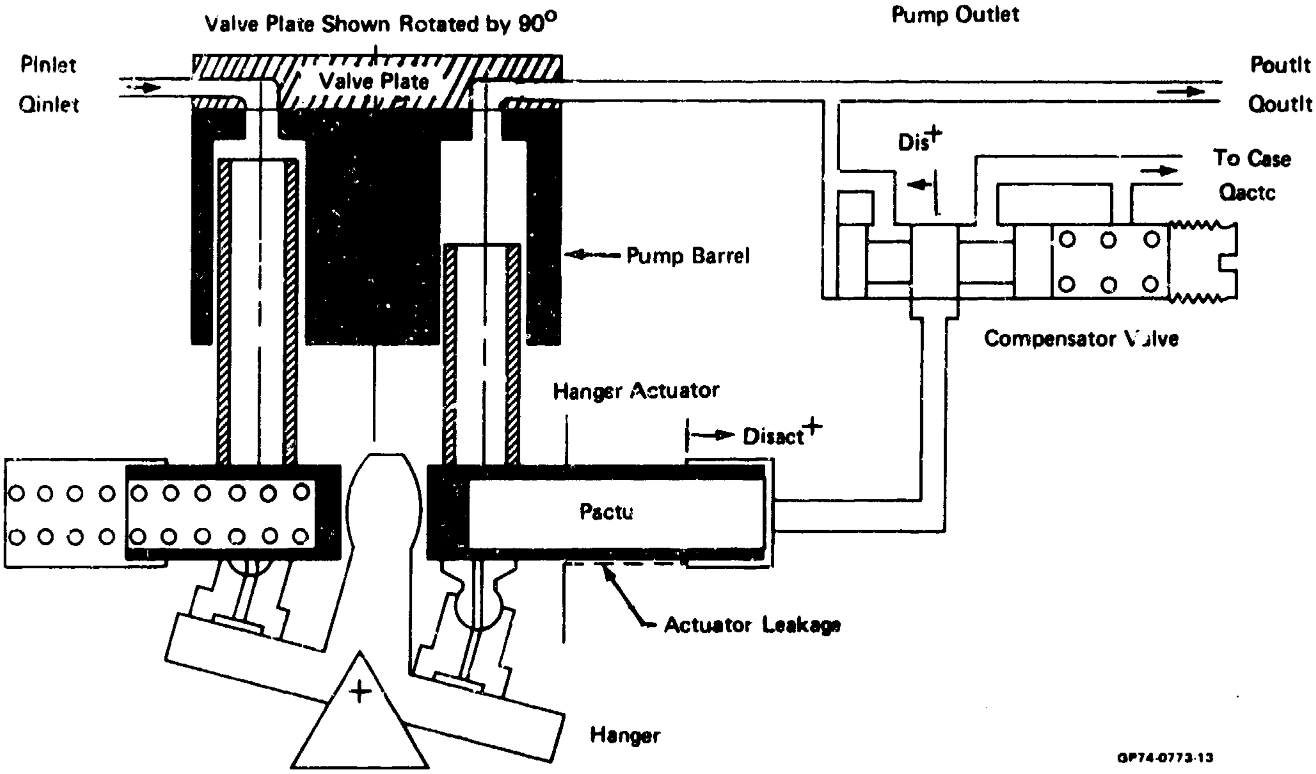


FIGURE 6.53-1
TYPE NO. 53 PRESSURE REGULATED VARIABLE DISPLACEMENT PUMP

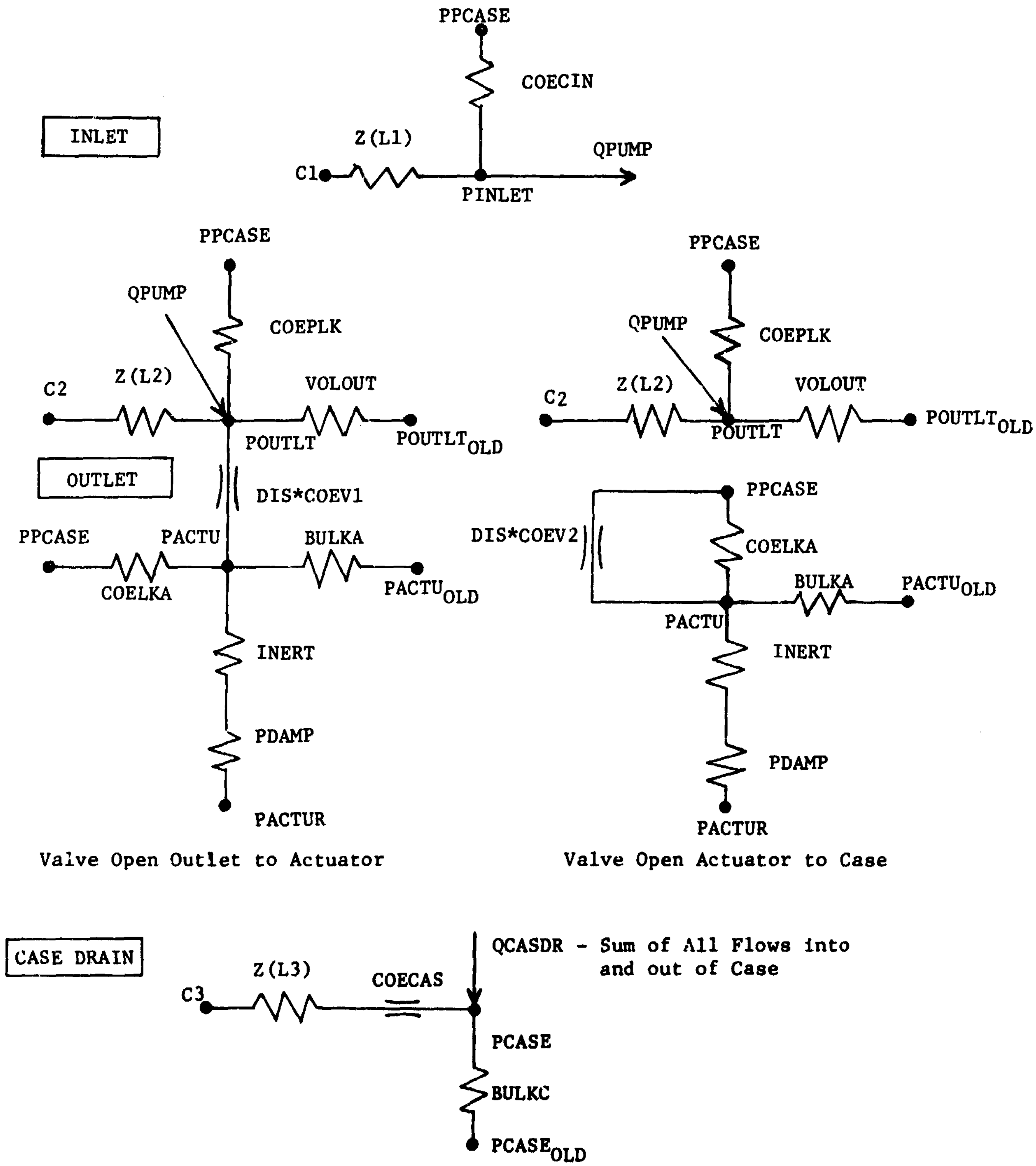


Figure 6.53-2
SCHEMATIC DIAGRAM FOR F-15 PUMP MODEL

The input data gives the actuator pressure due to the spring force at maximum flow, D(PSPRIM) and at zero flow, (D(PSPRIZ). P(PZRPM) is input as the pressure at zero actuator displacement and 3600 rpm. It is then modified to give the pressure at zero rpm by subtracting from it the slope of the pressure versus rpm curve, which is input as D(PSLRPM).

Using the F-15 pump experimental data and the actuator pressure predictions generated by the HSFR program, a formula was derived which related actuator pressure to the output pressure/bulk modulus ratio.

The test data collected at different fluid temperatures showed a need for temperature correction which was obtained via the bulk modulus. A correction factor DT(BULKO)/DT(POUTLT) is used where

DT(BULKO) = BULK(KTEMP(IND))*DT(DPVOA)/223000.

DT(DPVOA) is the reference pressure and 223000 is the reference bulk modulus of MIL-H-5606B at 3000 psi and 130°F.

The reference actuator pressure for this time step is then computed as

```
PACTUR = D(PSPRIZ)+PDISA*D(PSRIM) - VOLD*D(INERT)
         + PDISA*(D(PACCP)*DT(PRPM)**2 + D(PDISAC))
         + DT(PCASE) + (D(PZRPM) + D(PSLRPM)*DT(PRPM))
         *DT(BULKO)/DT(POUTLT)
```

where

PDISA = PREDICTED ACTUATOR POSITION BASED ON THE PREVIOUS POSITION AND VELOCITY

(=DT(DISACT)+DT(VELACT)*DELT)

VOLD = ACTUATOR VELOCITY (IN/SEC)

D(PACCP) = D(PACCP)/(D(DISAM)*3600**2)

D(PACCP) IS INPUT AS THE PRESSURE DUE TO PISTON ACCELERATION AT 3600 RPM AND MAXIMUM STROKE

D(PSPRIM) = (D(PSPRIM)-D(PSPRIZ))/D(DISAM)

D(PSPRIM) IS INPUT AS ACTUATOR PRESSURE DUE TO SPRING FORCE AT MAXIMUM PUMP DISPLACEMENT

D(PDISAC) = (D(PDISAC)-D(PZRPM))/D(DISAM)

D(PDISAC) IS INPUT AS ACTUATOR PRESSURE AT 3600 RPM AND MAXIMUM PUMP DISPLACEMENT

D(PZRPM) = D(PSRPM)-D(PSLRPM)*3600

D(PZRPM) IS INPUT AS ACTUATOR PRESSURE AT 3600 RPM AND ZERO PUMP DISPLACEMENT

The actuator pressure and damping characteristics are important variables since they govern how fast the pump goes from zero to full flow. No easy way exists to obtain these values accurately from purely dimensional data. The HSFR program is able to get within 30% of the measured actuator pressure and gives reasonable predictions of the variations with temperature and rpm. Measurements of actuator pressure and damping characteristics require a complex set of instrumentation and analysis of the data, to extract the variables would require inspired judgement. Therefore a reasonable initial actuator pressure D(PZRPM) which includes estimated values for the contributions of the spring is 800 psi. Other contributions to actuator pressure are input as zero.

Compensator Valve

The compensator valve position is assumed to be directly proportional to the differential pressure between outlet and case. The lag, due to the valve damping and hanger inertia is included in the computation of the actuator pressure as shown in the INERT and DAMP terms of Figure 6.51-2.

The valve spring rate D(INPPSI) is input in lbs/in and converted to in/psi,

D(INPPSI) = D(ARCOM)/D(INPPSI)

The differential pressure at which the valve opens from actuator to case D(DPVAC) is used to determine the valve position. Two pressures POUTMX and POUTMI are derived, POUTMX is the maximum outlet pressure that can be obtained assuming zero flow from the outlet to the actuator.

POUTMX = (C2/Z(L2) + QPUMP + DT(PCASE)*D(COEPLK) + D(VOLOUT)* OPOUT)*ZOUT

where

ZOUT = Z(L2)/(1.0 + Z(L2)*D(COEPLK) + Z(L2)*D(VOLOUT))

QPUMP = DT(QINLET) + QCASIN + QOSIN

The ZOUT term includes the impedance of the outlet volume (D(VOLOUT)).

The minimum pressure, POUTMI, which is the outlet pressure when the valve is just about to open to case is D(DPVAC).

The actual outlet pressure lies between these limits and is obtained by iteration. The valve orifice area is calculated for any position of the valve and the inlet port and is directly proportional to outlet pressure. The valve displacement is

DISOC = - D(INPPSI)*DT(POUTLT)+DT(COMVAL)

where DT(COMVAL) = D(VOVLP) + D(INPPSI)*D(DPVAC)

Flow From Outlet to Actuator

When DISOC is less than -D(VOVLP) the valve is open from the outlet to the actuator. A flow force displacement acts to close the valve and is computed as DT(PSIF) =D(FLOFRC)*DT(DIS)*(DT(POUTLT)-PACTUO)

Where DT(DIS) is the actual valve opening L = ABS(DISOC)-D(VOLVP))

The open slot width is

DT(DIS) = DT(DIS)-DT(PSIF)

The orifice areas is easily computed since the F-15 pump has square metering ports.

An orifice discharge coefficient is computed based on the diameter ratio between the equivalent diameter of the flow area to the slot width equivalent diameter and the Reynolds number of the previous actuator flow.

The computed actuator pressure and flow is based on outlet pressure, actuator leakage, damping inertia and actuator fluid compressibility as shown in Figure 6.53-2. When the actuator is bottomed the damping and inertia terms are ignored.

An iterative procedure is needed to compute the actuator pressure and flow. Since the F-15 pump is essentially a closed loop servo, a gain term was calculated as:

GAIN = 0.7/(1.+20.*S2ORHO(N)*D(INPPSI)*Z(L(@))*.65* D(WIDTH))

For the initial guess of outlet presure DT(POUTLT) is computed at DT(ITUP)*POUTMX + DT(MITUP)*POUTMI. (DT(ITUP)+GAIN, DT(MITUP)+1.-GAIN) This pressure is used to compute the valve position, area, and flow, DT(QACTU), into the actuator. The actuator flow is then used to recompute the pressure

TPOUT = PUTMX - DT(QACTU)*ZOUT

A check is made to see if the recomputed flow is within .05 psi of the pressure valve and if it is not, the outlet pressure is updated by

DT(POUTLT) = DT(POUTLT)*DT(MITUP) + TPOUT*DT(ITUP)

The choice of the DT(MITUP)/DT(ITUP) ratio and the initial guess ratio of DT(ITUP)/DT(MITUP) was made using an actual closed loop gain for the F-15 pump. For other pumps, adjustments of these ratios could reduce the number

<u>Closed Valve</u>

When the valve is within the overlap region, the valve flow and pressure are computed based on leakage due to diametrical clearance and overlap of the compensator valve.

Derivation of Equations for a Closed Valve

A schematic diagram of the leakage paths for a closed valve is shown in Figure 6.53-2. The actuator net flow is zero. The actuator pressure is determined from the diametral leakages past the valve, the actuator leakage, and a compressibility flow. A flow balance equation for the actuator pressure can be written as

$$DT(QACTU) = QCOMP + QLEAK + QCASE \qquad (1)$$

where

DT(QACTU) = flow from outlet to actuator (CIS)

QCOMP = compressibility flow (CIS)

QLEAK = actuator leakage flow (CIS)

QCASE = flow from actuator to case (CIS)

Each flow can be defined as:

$$DT(QACTU) = \frac{Db^3}{\rho\nu L 3.82}(DT(POUTLT) - DT(PACTU))$$

$$= A*(DT(POUTLT)-DT(PACTU))$$

where D = compensator radial distance (IN)

b = radial clearance (IN)

$\rho\nu$ = absolute viscosity $\frac{(lb\text{-}sec)}{in^2}$

L = lap distance (IN)

$$QCASE = \frac{Db^3}{\rho\nu L(D(VOVLP)-L)3.82}(DT(PACTU)-DT(PPCASE)) \qquad (2)$$

$$= B*(DT(PACTU) - DT(PPCASE))$$

where

D(VOVLP) = valve overlap (IN)

$$QLEAK = COELKA*(DT(PACTU) - DT(PPCASE)) \qquad (3)$$

$$QCOMP = DT(BULKA)*(DT(PACTU)-PACTUO) \tag{4}$$

Substituting equations (2), (3) and (4) into equation (1) after eliminating DT(PACTU) and grouping terms one can write:

$$DT(QACTU) = DT(POUTLT)*(DT(BULKA)+COELKA+B) - PACTUO*DT(BULKA)$$
$$- DT(PPCASE)*(COELKA + B) - DT\left(\frac{QACTU}{A}\right)(DT(BULKA)+COELKA+B)$$

Let CC = DT(BULKA) + COELKA+B

Solving for DT(QACTU)

$$DT(QACTU) = \frac{DT(POUTLT)*CC* - PACTUO*DT(BULKA)-DT(PPCASE)*(COELKA+B)}{1 + \frac{CC}{A}}$$

The actuator pressure can be written

$$DT(PACTU) = DT(POUTLT) - \frac{DT(QACTU)}{A}$$

Similarily the other flows can now be solved.

Flow From Actuator Piston to Case

When DISOC is greater than D(VOVLP) the valve opening is computed as

DT(DIS)= DISOC - D(VOVLP)

The opening is modified by the flow force and a discharge coefficient is computed. The valve is open allowing flow out of the actuator so that the pump flow, QPUMP increases. The valve displacement area and flow (DT(QACTC)) are calculated. Since the flow does not affect outlet pressure, no iteration is necessary.

The actuator leakage to case is assumed to be laminar since the passage is small around the actuator barrel. The actuator leakage is computed as

QACTLK = COELKA*(DT(PACTU) - DT(PPCASE))

with all the actuator flows known, the actuator velocity is calculated

DT(VELACT) = -FLOW/D(ARACT)

where flow has been determined from the valve positon and subsequent flow pressure balance. The new actuator position is then

DT(DISACT) = DT(DISACT) + (VOLD+DT(VELACT))*DELT/2.

A check is made to determine if the actuator is at the stroke limits. If it is the actuator flow must be recalculated. If the actuator is at maximum stroke (fully retracted with the pump at full stroke) then the actuator pressure drops to case pressure and DT(QACTC) is set to zero.

If the actuator is at minimum stroke (with the pump outlet flow negative), then DT(QACTU) and DT(PACTU) have to be recalculated so that the actuator leakage flow can be determined.

Pump Outlet Pressure and Flow

After the actuator pressure flows and valve outlet pressure DT(POUTLT) are computed, pump outlet flow is calculated as:

Q(L2) = -(QPUMP-QPLEAK-DT(QACTU)-(DT(POUTLT)-OPOUT)*D(VOLOUT))

Actual pump output pressure is then

P(L2) = C(L2)-Q(L2)*Z(L2)

Pump Case Outlet Pressure and Flow

The case outlet flow is determined using the schematic of Figure 6.51-2 to write a second degree equation for case drain flow:

DT(COECAS)*Q(L3)**2+(Z(L3)+DT(BULKC))*Q(L3)

-(DT(PCASE)+QCASDR*DT(BULKC)-C(L3))=0

where DT(PCASE) is the previous time step value of case pressure.

The above equation is solved for Q(L3) and the case pressure is computed as

DT(PCASE)=DT(PCASE)+(QCASDR+Q(L3))*DT(BULKC)

The outlet case pressure is then

P(L3) = C(L3)-Q(L3)*Z(L3)

6.53.2 Assumptions

The assumptions in a model of this nature are almost too many to enumerate. By its very nature the pump is a complex piece of equipment with multiple leak paths across the port plate, down the side of the piston and out of the shoes. The whole set of leak paths have been linearized and assumed to be constant for a constant output pressure, which is no doubt rather hard to accept. The alternative would be go into very detailed calculations with the leakage dependent on the piston load, hanger angle, RPM and anything else one could add. Unfortunately this too would probably be inaccurate so instead of an inaccurate complex leakage model we choose a simple leakage model, which could be improved when more data is available from the verification tests.

The forces on the hanger are not taken into account as it rotates only the hanger inertia and piston acceleration. Flow and leakage are all treated as though the pump had a continuous output rather than the individual pumping pistons.

In all the calculations the bulk modulus is treated as a constant for the high pressure (output) side and as a different constant for the low pressure (inlet) side, the elastic expansion of the volume cavities is included.

Friction effects have not been included primarily because of the cost of putting them in, but in actual fact the forces on the hanger have an oscillatory content which tend to keep it in motion. The pulsations of the outlet also tend to keep the valve in motion so that friction effects would not normally be significant.

6.53-3 Computation

The pump subroutine is set up using the HYTRAN program commons, plus the D, DT, DD, and L arrays.

1000 SECTION

In the 1000 section the constants are initialized where desirable for more efficient computation.

The remaining part of Section 1000 deals with the calculations of the steady state pump characteristics.

In order to balance the pump at some steady state condition, it is first necessary to establish what the pump characteristics are, over the maximum range of pump flow.

It was assumed that these characteristics could be approximated by a straight line interpolation between the pump conditions at maximum and minimum flows which correspond to zero and maximum actuator displacement respectively.

A chain of interdependent calculations are needed to derive the maximum and minimum conditions for:

DT(PACTU) = Actuator pressure

DT(QACTC) = Flow from actuator to case

DT(QACTU) = Flow from outlet to actuator through the valve

QACTLK = Actuator Leakage flow

DT(DISVLV) = Valve displacement

DT(POUTLT) = Valve chamber pressure which is the same as outlet pressure

The initial flows (QACTU and QACTC) are computed at the appropriate valve positions and this information is used in the steady state portion of the program.

1500 SECTION

Steady State Calculations

The pump which has three connections, has a node located at the inlet.

The leg which has the inlet connection receives the pump inlet pressure from the steady state routine.

For the leg which has the outlet connection as its first element, a value of the output flow ratio is computed as

$$DT(DISVLV) = \frac{QGUESS-DT(QOUTLT)}{DT(QINLET)-DT(QOUTLT)}$$

The ratio is calculated to determine the percentage of actuator flow that is actual leakage flow (QACTLK) into the pump case.

The output pressure rise is determine by the computed maximum outlet pressure at maximum valve displacement DT(POUTM) minus the pressure drop from case to inlet, DT(DELP13).

The outlet pressure rise is added to PQLEG(INEL, 5) and the output impedance (.00001) is added to PQLPG(INEL, 6).

DT(POUTLT) is initialized to DT(POUTM) - DT(DELP13)

And the outlet pressure PQLEG (INEL, 11) is also increased by DT(POUTM) - DT(DELP13).

LCS(INCL, 7) is set to 5 which means that the LEG formulae must be recalculated for every iteration because of the variation in inlet pressure.

The call for CON #3, the case drain, first gets the value of the flow guess for the case drain flow and then calculates the actuator leakage based

on the outlet flow ratio.

QACTLK = DT(DISVLV)*(DT(QACTC)-DT(QACTU))+DT(QACTU)

The pressure rise from inlet to case, DT(DELP13) is then calculated using the sum of the leakage flows divided by the coefficient of case to inlet leakage, D(COECIN). Since DT(DELP13) is based on a case drain flow, QCASDR/D(COECIN) is added along with DT(DELP13) to PQLEG (INEL, 5) for the constant pressure rise temperature.

1/D(COECIN), the case drain impedance is added to PQLEG(INEL, 6) and LCS (INEL, 7) is set to 5 so that the leg must be recalled each iteration since

PQLEG (INEL, 11) is increased by DT(DELP13) and DT(PCASE) is initialized to PQLEG (INEL, 11).

A test at the start of both the case drain and outlet calculations checks to see if INX = 1 which can only be true if CON 2 and 3 are the first or only elements in this leg.

The calculation method uses two interdependent pressure rises in two separate legs.

2000 SECTION

With the completion of the steady state calculations, where DT(PCASE), DT(POUTLT) and DT(PINLET) are initialized and a value for DT(DISVLV) is calculated, the pump variables can be initialized for the transient simulation.

The ratio DT(DISVLV) is used to initialize DT(DISACT) and DT(QACTU). Actuator velocity, DT(VELACT), valve displacement, DT(DISVLV), and DT(QACTC) are set to zero.

3000 SECTION

The 3000 section starts with a computation predicting the actuator displacement for the current time step. This value is used in the computation of the actuator pressure.

The pump inlet pressure is determined from the input impedance, Z(I,I), a pump inlet to case coefficient and pump flow as shown in the equivalent schematic diagram of the pump inlet model, Figure 6.53-2.

The next step is to determine the minimum (POUTMI) and maximum (POUTMX) pump output pressure range. The compensator valve displacement is then computed using an iterative technique as explained in the math model section. Once the valve position is known the actuator pressure and flow is computed along with the pump outlet pressure and flow.

The position of the valve, determines whether the actuator flow is going from the outlet to the actuator or from the actuator to case. The two equivalent circuit schematics used in the solution process are shown in Figure 6.53-2.

Figure 6.53-2 also shows the schematic for computing the case drain output pressure and flow. QCASDR is the sum of all flows into and out of the case:

QCASDR = QPLEAK + QACTLK + DT(QACTC) - D(ARACT)*DT(VELACT)

- QCASIN - QOSIN*2.

DT(BULKC) is the impedance of the case. It is computed as the fluid bulk modulus times DELT divided by the case volume.

The final values of outlet and case drain pressures and flows are passed to the appropriate P and Q arrays. The pump output horsepower is computed as:

POWER = -Q(L2)*(P(L1))/6600.0

A case pressure is computed using a simple integration.

DT(PPCASE)=2*DT(PCASE)-DT(PPCASE)

where

DT(PCASE)-LATEST CASE PRESSURE

DT(PPCASE)-PREDICTED CASE PRESSURE

If the pump inlet pressure is less than the minimum pump inlet pressure, a message is printed that gives the time at which pump cavitation occurs. The inlet flow is calculated using the minimum inlet pressure, D(PINMIN), instead of the inlet pressure, P(L1). The net flow is calculated and subtracted from the old cavity flow.

DT(QCAV) = DT(QCAV) - DT(QCNLET)-QCASIN

The value of DT(QCAV) is stored for the next time step and the cavitation model will control the inlet pressure until the pressure rises again and the flow refills the cavity.

6.53.4 Approximations

The approximation used in the program is the rather gross linearization between maximum and minimum flows used in the steady state calculations.

The remaining calculations follow the math model which is itself a large approximation.

6.53.5 Limitations

The current pump subroutine does not attempt to describe the true cavitation effects that can be caused by improper filling of the pistons. The effect on hanger angle and RPM vary greatly from pump to pump.

However, it is also a condition which the designer should avoid, by improving the pump inlet supply system, to prevent the inlet pressure dropping to the point where cavitation effects are a concern.

Another limitation, is the correct steady state prediction of pump outlet pressure, when the system flow exceeds the pump capacity. The transient section will limit the flows, but flow limitation is not included in the steady state section.

6.53.6 Variable Names

Name	Description	Dimension
A	Leakage coefficient from actuator to case	PSI/CIS**2
ACTRAD	Actuator Radious	IN
AEE	Computation Constant	-
APIS	Piston Area	IN**2
ALPHA	Computation constant	-
D(ARACT)	Actuator area	IN**2
D(ARCOM)	Compensator valve area	IN**2
AREA	Compensator valve flow area	IN**2
AREF,ASEC, ATHE,AL, B,BETA,BEE	Computation constants	-
DT(BULKA)	Impedance of actuator volume	PSI/CIS
DT(BULKC)	Impedance of case volume	PSI/CIS
DT(BULKO)	Oil bulk modulus	PSI
C,CHI,CC	Computation constants	-
D(COEALK)	Actuator flat depth	IN
D(COEALM)	Minimum actuator engagement	IN
DT(COEALS)	Dummy variable	-
DT(COEALZ)	Dummy variable	-
D(COECAS)	Constant term use to determine case outlet pressure drop	PSI/CIS**2
D(COECIN)	Coefficient of flow from case to inlet	CIS/PSI
COELKA	Dummy varirble	-
D(COEOSO)	Coefficient of outlet flow due to actuator motion	CIS/(IN/SEC)
D(COEPLK)	Coefficient of pump leakage (outlet to case)	CIS/PSI
DT(COMVAL)	Computation constant	-

Name	Description	Dimension
CONA,CONB	Dummy variables	-
C1	Inlet characteristic pressure	PSI
C2	Outlet characteristic pressure	PSI
C3	Case drain characteristic pressure	PSI
DT(COM1), DEE	Computation variable	-
DT(DELP13)	Pressure drop from case to inlet	PSI
DT(DELP23)	Pressure drop from outlet to case	PSI
DT(DIS)	Valve opening	IN
DT(DISACT)	Actuator displacement	IN
D(DISAM)	Actuator position at maximum pump displacement	IN
D(DISAMI)	Actuator position at minimum pump displacement	IN
D(DISP)	Theoretical maximum pump displacement changed to IN**3/IN/RPM	IN**3/REV
DT(DISVLV)	Valve displacement	IN
D(DISVM)	Maximum valve displacement	IN
DPDAMP, DT(DOPEN)	Dummy variable	-
D(DPVAC)	Pressure at which valve is open from outlet to actuator	PSI
DT(DPVOA)	Pressure at which valve starts to open from outlet to actuator	PSI
DP1,DP2, DRATIO, DT(REF)	Dummy variable	-
FACTC	Previous flow from actuator to case	CIS
FACTU	Previous flow from outlet to actuator	CIS
D(FLOFRC)	Flow force on valve spool constant	-
FLOW	Net actuator flow	CIS

Name	Description	Dimension
GAIN	Closed loop pump gain	-
D(INERT)	Hanger mass	-
D(INPPSI)	Spring rate of spool changed to IN/PSI	LB/IN
DT(ITUP), DT(MITUP)	Iteration constants	-
D(LEVACT)	Actuator lever arm at midstroke	IN
OPOUT	Previous value of outlet pressure	PSI
DT(ORF)	Valve discharge coefficient	-
D(PACCP)	Actuator pressure due to piston acceleration at 3600 rpm & maximum pump displacement	PSI
DT(PACTU)	Actuator pressure	PSI
PACTUO	Previous value of actuator pressure	PSI
DT(PACTUR)	Reference actuator pressure	PSI
DT(PCASE)	Case pressure	PSI
D(PDAMP)	Hanger damping	PSI/IN/SEC
PDISA	Predicted actuator displacement	IN
D(PDISAC)	Pressure at 3600 RPM and maximum pump displacement: changed to rate of change of pressure with actuator position	PSI
DT(PINLET)	Inlet pressure	PSI
D(PINMIM)	Outlet pressure	PSI
POUTM, POUTMI POUTMX, DT(POUTM)	Dummy variables	-
POWER	Pump output horsepower	HP
DT(PPCASE)	Predicted case pressure	PSI
DT(PPOWER)	Output horsepower	HP
DT(PRPM)	Pump operating speed	RPM
DT(PSIF)	Valve displacement due to flow force	IN

Name	Description	Dimension
D(PSLRPM)	Slope of pressure vs RPM curve	PSI/RPM
D(PSPEED)	Pump operating speed	RPM
D(PSPRIM)	Actuator pressure due to spring force maximum pump displacement adjusted to slope PSI/IN	PSI
D(SPRIZ)	Actuator pressure due to spring force at zero pump displacement	PSI
D(PZRPM)	Actuator pressure inputted at 3600 RPM and zero pump displacement; adjusted to zero RPM	PSI
DT(QACTC)	Actuator flow with valve is open from actuator to case	CIS
QACTLK	Leakage flow from actuator to case	CIS
DT(QACTU)	Actuator flow when valve is open from outlet to actuator	CIS
QCASDR	Sum of all flows into and out of the case	CIS
QCASIN	Flow from case to inlet	CIS
DT(QCAV)	Temporary variable	CIS
DT(QINLET)	Inlet flow	CIS
QOSIN	Outlet flow due to actuator motion	CIS
DT(QOUTLT)	Outlet flow	CIS
QPLEAK	Pump leakage flow	CIS
QPUMP	Pump flow	CIS
REN	Reynolds number	-
TPOUT	Dummy variable	-
DT(VELACT)	Actuator velocity	IN/SEC
D(VLVOL)	Valve overlap	IN
D(VOLACT)	Actuator volume	IN**3
D(VOLCAS)	Case volume	IN**3
VOLD	Previous actuator velocity	IN/SEC

Name	Description	Dimension
D(VOLOUT)	Outlet volume	IN**3
D(VOVLP)	Valve overlap	IN
D(WIDTH)	Slot width	IN
ZOUT	Outlet impedance	PSI/CIS

6.53.7 Subroutine Listing

```
      SUBROUTINE PUMP53 (D,DT,DD,L)
C **** REVISED OCTOBER 1979 ****
C       PUMP51 WITH REWORKED INPUT DATA REQUIREMENTS
C       CDC FILE NAME :  RPUMP53
      COMMON NTELPL,NTOLPL,IPT,IPOINT,NPTS,INEL,KNEL,NTOPL,NLPLT(61,3),
     1 PQLEG(90,12),LCS(90,10),ILEG(1400),PN(90),QN(90)
      COMMON/SUB/PARM(150,9),PM(1500),QM(1500),P(300),Q(300),C(300)
     1,Z(300),RHO(20),S2ORHO(20),VISC(20),BULK(20),TEMP(20),PVAP(20)
     2,ATPRES,T,DELT,TFINAL,PLTDEL,PI,TITLE(20),LEGN,ICON
     3,KTEMP(99),LSTART(150),NLPT(150),LTYPE(99),NC(99),INX,INZ
     4,INV,ISTEP,NLINE,NEL,IND,IENTR,MNLINE,MNEL,MNLEG,MNNODE,MNPLOT
     5,MNLPTS,MDS
      DIMENSION D(37),DT(43),DD(1),L(10)
      INTEGER ARCOM,WIDTH,FLOFRC,APIS,LEVACT,ARACT,PSPRIZ,PSPRIM,
     1 PACCP,PZRPM,PSLRPM,PDISAC,PDAMP,DISP,DPVOA,DPVAC,ORF,
     2 DISAM,DISAMI,COEALK,COEALM,COEPLK,COECIN,VOLCAS,PINMIN,
     3 PSPEED,PRPM,PPOWER,QACTU,QACTC,PACTU,POUTLT,DIS,PSIF,
     4 PCASE,PINLET,DREF,DOPEN,BULKC,VELACT,DISACT,DISVLV,AREF,
     5 COEALZ,COMVAL,DELP13,DELP23,COECAS,POUTM,BULKO,COEOSO
     6 ,QINLET,QOUTLT,DISVM,VOVLP,VOLACT,BULKA,VOLOUT,PPCASE
     7 ,DCOM1,PACTUR,QCAV
C     D( ) ARRAY *****
      DATA DPVAC/1/,INPPSI/2/,ARCOM/3/,WIDTH/4/,FLOFRC/5/,VOVLP/6/,
     1 APIS/7/,LEVACT/8/,ARACT/9/,PSPRIZ/10/,PSPRIM/11/,
     2 PACCP/12/,PZRPM/13/,PDISAC/14/,PSLRPM/15/,PDAMP/16/,
     3 DISP/17/,DISAM/18/,DISAMI/19/,COEALK/20/,COEALM/21/,
     4 COEPLK/22/,COECIN/23/,VOLCAS/24/,PINMIN/25/,PSPEED/26/,
     5 COEOSO/27/,DISVM/28/,COECAS/29/,INERT/30/,VOLACT/31/,VOLOUT/32/
C     DT( ) ARRAY *****
      DATA PRPM/1/,PPOWER/2/,ITEM/3/,QACTU/4/,QACTC/5/,
     1 PACTU/7/,POUTLT/8/,PCASE/9/,PINLET/10/,DREF/11/,DOPEN/12/,
     2 BULKC/13/,VELACT/14/,DISACT/15/,DISVLV/16/,PSIF/17/,COEALZ/18/,
     3 COMVAL/19/,DPVOA/20/,POUTM/21/,DELP13/22/,DELP23/23/,BULKO/24/
     4 ,QINLET/25/,QOUTLT/26/,ITUP/27/,MITUP/28/,BULKA/29/,PPCASE/30/
     5 ,ORF/31/,AREF/32/,DIS/33/,PACTUR/34/,DCOM1/35/,QCAV/36/
C
      IF(IENTR) 1000,2000,3000
C *** 1000 SECTION
 1000 CONTINUE
      IF (INEL.NE.0) GO TO 1500
      DO 1001 I=1,43
 1001 DT(I)=0.0
      N=KTEMP(IND)
      IF(N.LT.11) N=N+10
      DT(BULKC)=(BULK(N)*DELT)/D(VOLCAS)
      DT(BULKA)=D(VOLACT)/(BULK(N)*DELT)
      D(VOLOUT)=D(VOLOUT)/(BULK(N)*DELT)
      D(COEOSO)=D(COEOSO)*D(APIS)*L(4)*.5/D(LEVACT)
      DT(AREF)=D(WIDTH)*D(WIDTH)
      DT(DREF)=SQRT(DT(AREF)*4./PI)
      DT(DREF)=DT(DREF)/DT(AREF)
```

6.53.7 (Continued)

```
 3217 TPOUT=POUTMX-DT(QACTU)*ZOUT
      IF(ABS(TPOUT-DT(POUTLT)).LT.0.05) GO TO 3230
      DT(POUTLT)=DT(POUTLT)*DT(MITUP)+TPOUT*DT(ITUP)
      ICOUNT=ICOUNT+1
      IF(ICOUNT.EQ.25)WRITE(6,999)ICOUNT
      IF(ICOUNT.EQ.25)WRITE(6,998)DT(POUTLT),TPOUT,POUTMX,POUTMI
 998  FORMAT(10X,4F20.5)
      IF(ICOUNT.EQ.25)GO TO 3230
 999  FORMAT(10X,13HEXCEEDED ITER,I10)
      GO TO 3210
C
C     FLOW FROM ACTUATOR PISTON TO CASE
C
 3220 CONTINUE
      DT(PSIF)=D(FLOFRC)*DT(DIS)*(PACTUO-DT(PPCASE))
      DT(DIS)=DISOC-D(VOVLP)
      DT(DIS)=DT(DIS)-DT(PSIF)
      IF(DT(DIS).LE.0.0)GO TO 3222
      IF(DT(DIS).GT.D(DISVM))DT(DIS)=D(DISVM)
C   COMPUTE FLOW AREA
      AREA=(DT(DIS))*D(WIDTH)
      DT(DOPEN)=SQRT(AREA*4./PI)
      DRATIO=DT(DOPEN)/DT(DREF)
      REN=FACTC*DT(DREF)/VISC(N)
      CALL ORCOEFF(REN,DRATIO,DT(ORF))
      A=AREA*DT(ORF)*S2ORHO(N)
      DT(QACTC)=-B*A**2/2.+A/2.*SQRT((A*B)**2+4*ABS(DP2))
      FLOW=-DT(QACTC)
      DT(PACTU)=DT(PPCASE)+(DT(QACTC)/A)**2
 3230 IF(POWER.NE.0.0) GO TO 3250
C
C     TEST ACTUATOR DISPLACEMENT AGAINST MAXIMUM STROKE
C
      QACTLK=(DT(PACTU)-DT(PPCASE))*COELKA
C     FLOW + OUTLET TO ACT,ACT TO MIN STROKE
C     FLOW - ACT TO CASE,ACT TO MAX STROKE
      DT(VELACT)=-FLOW/D(ARACT)
      DT(DISACT)=DT(DISACT)+(VOLD+DT(VELACT))/2.*DELT
      CALL XLIMIT(DT(DISACT),DT(VELACT),POWER,D(DISAMI),D(DISAM))
C
C     POWER    -             DESCRIPTION
C       +1          ACTUATOR AT MAX STROKE(PUMP FULL FLOW)
C        0          ACTUATOR WITHIN TRAVEL LIMITS
C       -1          ACTUATOR AT MIN STROKE(ZERO PUMP FLOW)
C
```

6.53.7 (Continued)

```
      D(COECAS)=RHO(N)*1.3888888/(D(COECAS)*D(COECAS))
      DT(DCOM1)=SQRT(D(ARCOM)*4/PI)
      DT(DCOM1)=DT(DCOM1)*1.E-12/(RHO(N)*VISC(N)*3.82)
      ACTRAD=SQRT(D(ARACT)*4./PI)/2.
      DEE=ACTRAD-D(COEALK)
      OPOUT=2.*SQRT(ACTRAD**2-DEE**2)
      ATRI=.5*DEE*OPOUT
      AEE=DEE/ACTRAD
      ATHE=ACTRAD*ACTRAD*ACOS(AEE)
      ASEC=ATHE-ATRI
      BEE=ASEC/OPOUT
      DT(COEALZ)=(OPOUT*(BEE**3))/(6.*VISC(N)*RHO(N))
      D(INERT)=D(INERT)/(DELT*D(ARACT))
      D(FLOFRC)=.43*D(WIDTH)/D(INPPSI)
      D(INPPSI)=D(ARCOM)/D(INPPSI)
      DT(COMVAL)=D(VOVLP)+D(INPPSI)*D(DPVAC)
      D(VOVLP)=D(VOVLP)/2.
      D(DISP)=D(DISP)/60./D(DISAM)
      D(PSPRIM)=(D(PSPRIM)-D(PSPRIZ))/D(DISAM)
      D(PACCP)=D(PACCP)/(D(DISAM)*3600.**2)
      D(PDISAC)=(D(PDISAC)-D(PZRPM))/D(DISAM)
      D(PZRPM)=D(PZRPM)-D(PSLRPM)*3600.
      DT(PRPM)=D(PSPEED)
      DT(DPVOA)=(D(VOVLP)+DT(COMVAL))/D(INPPSI)
      DT(BULKO)=BULK(N)*DT(DPVOA)/223000.
      DT(DISACT)=D(DISAM)*.5
C***  THIS SECTION CALCULATES THE STEADY STATE CHARCTERISTICS
C     OF THE PUMP OUTPUT PRESSURE VERSUS FLOW.
      COELKA=DT(COEALZ)/D(COEALM)
      DT(POUTLT)=D(DPVAC)+(DT(DPVOA)-D(DPVAC))*.5
 1260 DT(QACTC)=DT(QACTU)
 1270 DT(PACTU)=D(PSPRIZ)+DT(DISACT)*D(PSPRIM)+0.0
     1 +DT(DISACT)*(D(PACCP)*DT(PRPM)**2+D(PDISAC))
     2 +(D(PZRPM)+D(PSLRPM)*DT(PRPM))*DT(BULKO)/DT(POUTLT)
      IF(IENTR.EQ.0)RETURN
      DT(QACTU)=COELKA*DT(PACTU)
      DT(DISACT)=D(DISAM)
      IF(DT(QACTC).EQ.0.)GO TO 1260
      DT(POUTM)=DT(POUTLT)
      DT(QINLET)=D(DISP)*DT(PRPM)*D(DISAM)-DT(POUTM)*D(COEPLK)
     1 -DT(QACTU)
      DT(QOUTLT)=-(D(COEPLK))*DT(POUTLT)-DT(QACTC)
      DT(DELP23)=-DT(QOUTLT)/(DT(QINLET)-DT(QOUTLT))
      DT(DISVLV)=.5
      DT(DELP13)=10.
      GAIN=.7/(1.+20.*S2ORHO(N)*D(INPPSI)*Z(L(2))*.65*D(WIDTH))
      DT(ITUP)=GAIN
      DT(MITUP)=1.-GAIN
      DT(PACTUR)=DT(PACTU)
      RETURN
```

6.53.7 (Continued)

```
C
C      STEADY STATE CALCULATION SECTION
C     IND=COMPONENT #,KNEL=CONNECTION #,INEL=LEG #
C     CON #1=INLET,CON #2=OUTLET,CON #3=CASE DRAIN
C     THE INLET IS A NODAL POINT IN THE SYSTEM
C
 1500 IF(KNEL-2)1510,1530,1520
 1510 DT(PINLET)=PQLEG(INEL,11)
      GO TO 1600
 1520 IF(INX.NE.1) GO TO 1700
      QCASDR=PQLEG(INEL,1)*PQLEG(INEL,2)
      DT(6)=QCASDR
      QACTLK=DT(DISVLV)*(DT(QACTC)-DT(QACTU))+DT(QACTU)
      DT(DELP13)=(DT(POUTM)*D(COEPLK)+QACTLK-QCASDR)/D(COECIN)
      PQLEG(INEL,5)=PQLEG(INEL,5)+DT(DELP13)+QCASDR/D(COECIN)
      PQLEG(INEL,6)=PQLEG(INEL,6)+1.0/D(COECIN)
      LCS(INEL,7)=5
      PQLEG(INEL,11)=PQLEG(INEL,11)+DT(DELP13)
      DT(PCASE)=PQLEG(INEL,11)
      PQLEG(INEL,8)=PQLEG(INEL,8)+D(COECAS)
      PQLEG(INEL,11)=PQLEG(INEL,11)-D(COECAS)*(QCASDR**2)
      GO TO 1600
 1530 IF(INX.NE.1) GO TO 1700
      DT(DISVLV)=PQLEG(INEL,1)*PQLEG(INEL,2)-DT(QOUTLT)
      DT(DISVLV)=DT(DISVLV)/(DT(QINLET)-DT(QOUTLT))
      IF(DT(DISVLV).LT.0.0) DT(DISVLV)=0.0
      IF(DT(DISVLV).GT.1.0) DT(DISVLV)=1.0
      PQLEG(INEL,5)=PQLEG(INEL,5)+DT(POUTM)-DT(DELP13)
      PQLEG(INEL,6)=PQLEG(INEL,6)+0.00001
      PQLEG(INEL,11)=PQLEG(INEL,11)+DT(POUTM)-DT(DELP13)
      DT(POUTLT)=PQLEG(INEL,11)
      LCS(INEL,7)=5
 1600 RETURN
 1700 WRITE(6,1800) IND,KNEL,INEL
 1800 FORMAT(5X,46H CALL SEQUENCE ERROR DETECTED IN COMPONENT NO     ,
     1 I5,14H CONNECTION NO  ,I5,7H LEG NO  ,I5)
      WRITE(6,943)
  943 FORMAT(10X,33HPROGRAM STOP IN SUBROUTINE PUMP51)
      STOP 6054
C *** 2000 SECTION
 2000 CONTINUE
      DT(DISACT)=DT(DISVLV)*D(DISAM)
      DT(QACTU)=DT(DISVLV)*(DT(QACTC)-DT(QACTU))+DT(QACTU)
      DT(QACTC)=0.0
      DT(DISVLV)=0.0
      DT(PPCASE)=DT(PCASE)
C     WRITE(6,2010)(DT(I),I=1,40)
 2010 FORMAT(1X,10E12.5)
      GO TO 1270
```

6.53.7 (Continued)

```
C
C *** 3000 SECTION
 3000 CONTINUE
      N=KTEMP(IND)
      ICOUNT=0
C
C     CALCULATE TRANSIENT RESPONSE OF PUMP
C
      POWER=0.0
      L1=L(1)
      L2=L(2)
      L3=L(3)
      C1=C(L1)
      C2=C(L2)
      C3=C(L3)
      VOLD=DT(VELACT)
      DPDAMP=D(PDAMP)
      FACTU=ABS(DT(QACTU))
      FACTC=ABS(DT(QACTC))
      DPDAMP=D(ARACT)/(DPDAMP+D(INERT))
      DT(QACTU)=0.0
      DT(QACTC)=0.0
      PACTUO=DT(PACTU)
      PDISA=DT(DISACT)+DT(VELACT)*DELT
C
C     COMPUTE REFERENCE ACTUATOR PRESSURE
C
      DT(PACTUR)=D(PSPRIZ)+PDISA*D(PSPRIM)+VOLD*D(INERT)
     1 +PDISA*(D(PACCP)*DT(PRPM)**2+D(PDISAC))+0.0
     2 +(D(PZRPM)+D(PSLRPM)*DT(PRPM))*DT(BULKO)/DT(POUTLT)
      QOSIN=-VOLD*D(COEOSO)
      QPUMP=PDISA*D(DISP)*DT(PRPM)+QOSIN
      DT(PINLET)=(C1/Z(L1)+DT(PPCASE)*D(COECIN)-QPUMP)
     1  /(1.0/Z(L1)+D(COECIN))
      IF(DT(PINLET).LT.D(PINMIN).OR.DT(QCAV).GT.0.0)GO TO 3100
      DT(QINLET)=(C1-DT(PINLET))/Z(L1)
      QCASIN=(DT(PPCASE)-DT(PINLET))*D(COECIN)
      GO TO 3110
 3100 DT(PINLET)=D(PINMIN)
      DT(QINLET)=(C1-DT(PINLET))/Z(L1)
      QCASIN=(DT(PPCASE)-DT(PINLET))*D(COECIN)
      DT(QCAV)=DT(QCAV)-DT(QINLET)-QCASIN
      IF(DT(QCAV).LT.0.0)DT(QCAV)=0.0
 3110 QPUMP=DT(QINLET)+QCASIN+QOSIN
      COELKA=DT(COEALZ)/(D(COEALM)+0.0)
      ZOUT=Z(L2)/(1.0+Z(L2)*D(COEPLK)+Z(L2)*D(VOLOUT))
      CPOUT=DT(POUTLT)
      B=1./(DPDAMP+DT(BULKA)+COELKA)
      DP1=(DT(PPCASE)*COELKA+PACTUO*DT(BULKA)+DT(PACTUR)*DPDAMP)*B
      DP2=B*(DT(PACTUR)*DPDAMP+PACTUO*DT(BULKA)+DT(PPCASE)*(COELKA
     + -1./B))
```

6.53.7 (Continued)

```
C
C      DETERMINE COMPENSATOR POSITION
C
 3200 POUTMX=(C2/Z(L2)+QPUMP+DT(PPCASE)*D(COEPLK)+D(VOLOUT)*OPOUT)*ZOUT
      POUTMI=D(DPVAC)
      DT(POUTLT)=POUTMX
 3210 CONTINUE
C     DISOC=-7.86357E-5*DT(POUTLT)+.229569
      DISOC=-D(INPPSI)*DT(POUTLT)+DT(COMVAL)
      IF(DISOC.GE.-D(VOVLP))GO TO 3221
C     VALVE OPEN OUTLET TO ACTUATOR
      DT(PSIF)=D(FLOFRC)*DT(DIS)*(DT(POUTLT)-PACTUO)
      DT(DIS)=ABS(DISOC)-D(VOVLP)
      DT(DIS)=DT(DIS)-DT(PSIF)
      IF(DT(DIS).LE.0.0)GO TO 3222
      IF(DT(DIS).GT.D(DISVM))DT(DIS)=D(DISVM)
C     COMPUTE FLOW AREA
      AREA=(DT(DIS))*D(WIDTH)
      DT(DOPEN)=SQRT(AREA*4./PI)
      DRATIO=DT(DOPEN)/DT(DREF)
      REN=FACTU*DT(DREF)/VISC(N)
      CALL ORCOEFF(REN,DRATIO.DT(ORF))
      IF(POWER.EQ.0.0)GO TO 3216
C     VALVE OPENED TO PRESSURE - ACT BOTTOMED
      CONA=(AREA*DT(ORF)*S2ORHO(N))**2/(2.*COELKA)
      CONB=2.*ABS(DT(POUTLT)-DT(PPCASE))*COELKA/CONA
      DT(QACTU)=CONA*(SQRT(1.+CONB)-1.)
      FLOW=DT(QACTU)
      DT(PACTU)=DT(PPCASE)+DT(QACTU)/COELKA
      GO TO 3217
 3221 IF(DISOC.GT.D(VOVLP))GO TO 3220
C     VALVE IN LAPPED REGION
 3222 DT(DIS)=0.0
      POVLP=ABS(DISOC+D(VOVLP))+.00001
      POVLC=ABS(DISOC-D(VOVLP))+.00001
      A1=DT(DCOM1)/POVLP
      B1=DT(DCOM1)/POVLC
      CC=DT(BULKA)+COELKA+B1+DPDAMP
      DT(QACTU)=DT(POUTLT)*CC-PACTUO*DT(BULKA)-DT(PPCASE)*(COELKA+B1)
      DT(QACTU)=DT(QACTU)-DT(PACTUR)*DPDAMP
      DT(QACTU)=DT(QACTU)/(1.+CC/A1)
      DT(PACTU)=DT(POUTLT)-DT(QACTU)/A1
      DT(QACTC)=B1*(DT(PACTU)-DT(PPCASE))
      FLOW=0.0
      GO TO 3217
 3216 CONTINUE
C     VALVE OPENED TO PRESSURE - ACT NOT BOTTOMED
      A=AREA*DT(ORF)*S2ORHO(N)
      DT(QACTU)=-B*A**2/2.+A/2.*SQRT((A*B)**2+4*ABS(DT(POUTLT)-DP1))
      DT(PACTU)=DT(POUTLT)-(DT(QACTU)/A)**2
      FLOW=DT(QACTU)
```

6.53.7 (Continued)

```
      IF(POWER.EQ.0.0) GO TO 3300
      IF(DT(PINLET).LE.D(PINMIN)) GO TO 3240
      QPUMP=D(DISP)*DT(PRPM)*DT(DISACT)
      QOSIN=0.0
      ICOUNT=1
 3240 IF(POWER.EQ.-1.0) GO TO 3200
      DT(QACTC)=0.0
      DT(PACTU)=DT(PPCASE)
      QACTLK=0.0
      GO TO 3300
 3250 CONTINUE
      QACTLK=DT(QACTU)
 3300 P(L1)=DT(PINLET)
      Q(L1)=DT(QINLET)
      QPLEAK=(DT(POUTLT)-DT(PPCASE))*D(COEPLK)
      Q(L2)=-(QPUMP-QPLEAK-DT(QACTU)-(DT(POUTLT)-OPOUT)*D(VOLOUT))
      DT(QOUTLT)=-Q(L2)
      DT(DISVLV)=(DT(POUTLT)-DT(PPCASE)-D(DPVAC))*D(INPPSI)
      DT(POUTLT)=C(L2)-Q(L2)*Z(L2)
      P(L2)=DT(POUTLT)
      QCASDR=QPLEAK+QACTLK+DT(QACTC)-D(ARACT)*DT(VELACT)-QCASIN-QOSIN*2.
      DT(6)=QCASDR
      ALPHA=D(COECAS)
      BETA=Z(L3)+DT(BULKC)
      CHI=DT(PCASE)+QCASDR*DT(BULKC)-C(L3)
      Q(L3)=(-BETA+SQRT(BETA**2+4.*ALPHA*ABS(CHI)))/(2.*ALPHA)
      Q(L3)=SIGN(Q(L3),-CHI)
      DT(PPCASE)=DT(PCASE)
      DT(PCASE)=DT(PCASE)+(QCASDR+Q(L3))*DT(BULKC)
      P(L3)=C(L3)-Q(L3)*Z(L3)
      DT(PPCASE)=2.*DT(PCASE)-DT(PPCASE)
C
      POWER=-Q(L2)*(P(L2)-P(L1))/6600.0
      DT(PPOWER)=POWER
      IF(DT(PINLET).LE.D(PINMIN)) WRITE(6,3310)T,IND
      RETURN
 3310 FORMAT(2X,36H******** PUMP CAVITATION ONSET AT T= ,E12.5,
     +10X,8HCOMP NO.,I5)
      END
```

6.21 TYPE #21 TWO WAY CONTROL VALVE

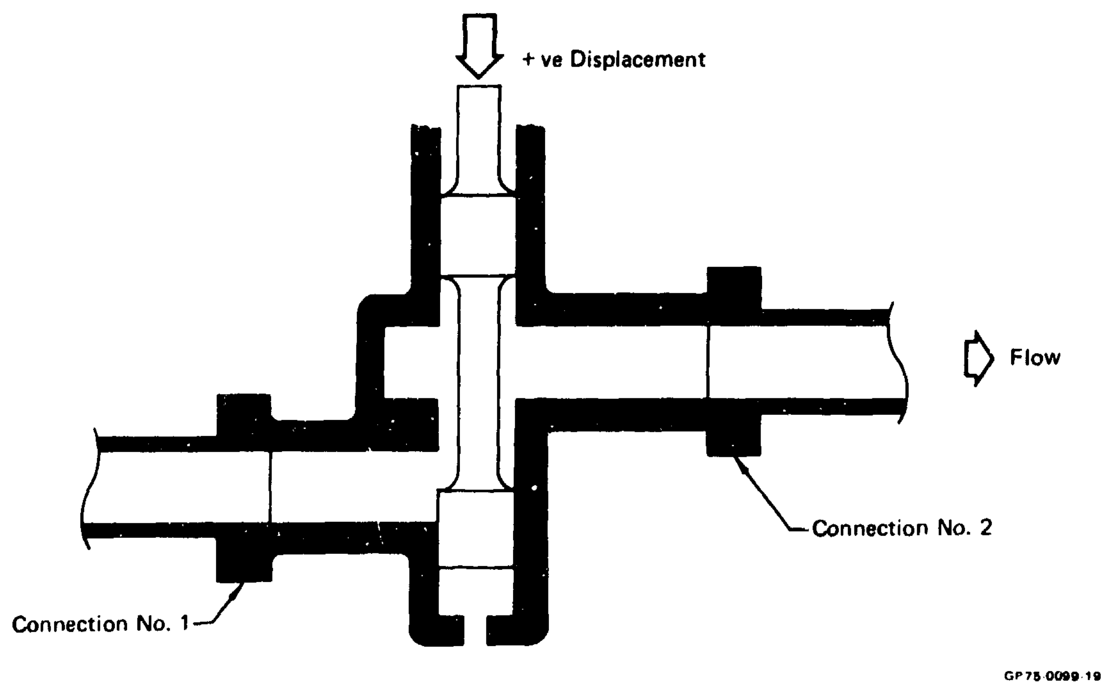


FIGURE 6.21-1
TYPE NO. 21 TWO-WAY VALVE

The type #21 valve uses an externally controlled time history input. The valve opening versus time is derived from the input data tabulated on the third and fourth data cards.

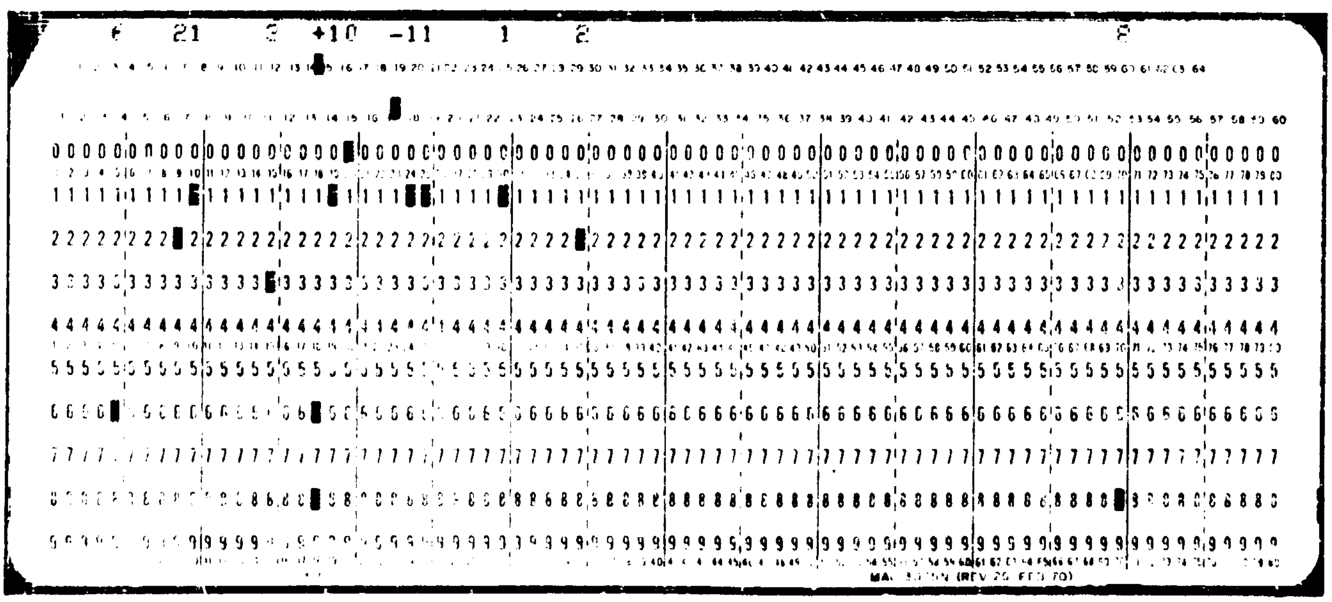
Valve sizing can be input either as rated flow and pressure drop or as physical dimensions (slot width and discharge coefficient).

The total number of values in the time/position table must be equal to the number input in column 70 of the first card.

CARD NUMBER 1

COLUMN	FORMAT	DATA
1-5	I5	Component Number
6-10	I5	Type Number = 21
11-15	I5	Number of Real Data Cards = 3 or more
16-20	I5	Line Number (with sign) attached to Connection 1
21-25	I5	Line Number (with sign) attached to Connection 2
26-30	I5	Type of data input (1=rated flow, 0 or default=valve slot)
31-35	I5	Rating fluid (1=MIL-H-5606, 2=MIL-H-83282, 3=Skydrol 500)
36-40	I5	
41-45	I5	
46-50	I5	
51-55	I5	
56-60	I5	
61-65	I5	
66-70	I5	Number of data points in table
71-75	I5	
76-80	I5	Temperature/Pressure Code (See Page 4.0-2)

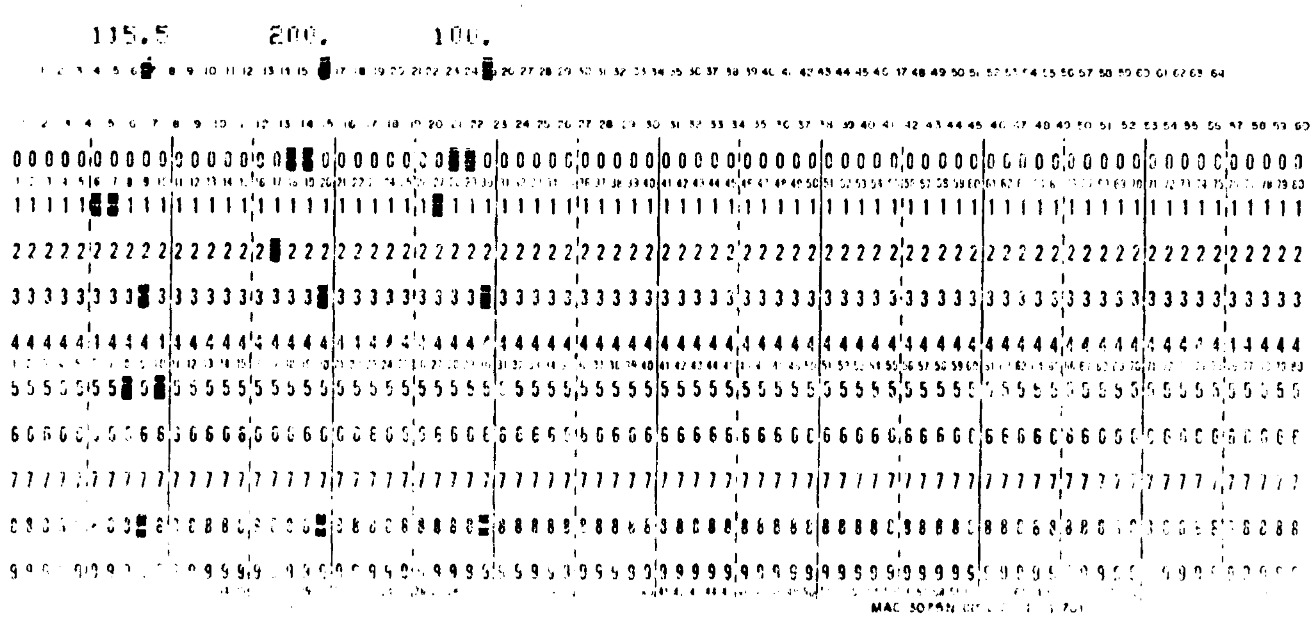
EXAMPLE CARD



CARD NUMBER 2

COLUMN	FORMAT	DATA	DIMENSIONS
1-10	E10.0	Valve slot width or rated flow	IN or CIS
11-20	E10.0	Discharge coefficient or rated DP	- or PSI
21-30	E10.0	Rating fluid temperature	DEGREES F
31-40	E10.0		
41-50	E10.0		
51-60	E10.0		
61-70	E10.0		
71-80	E10.0		

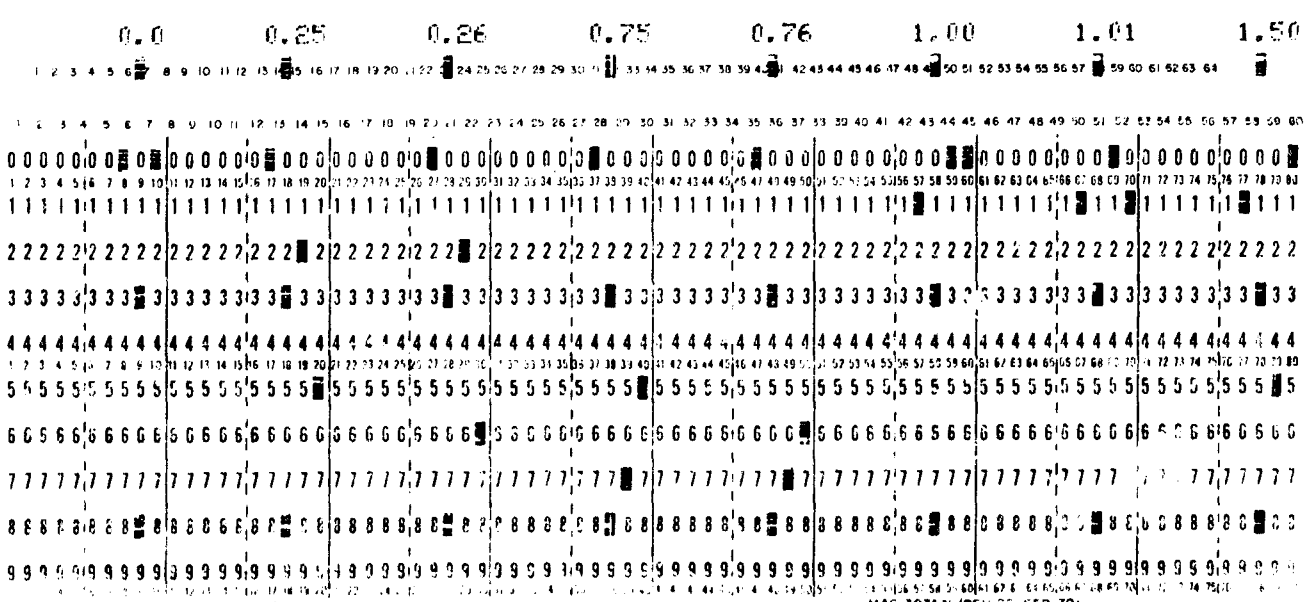
EXAMPLE CARD



CARD NUMBER 3

COLUMN	FORMAT	DATA	DIMENSIONS
1-10	E10.0	First time value (should = 0)	SEC
11-20	E10.0		
21-30	E10.0	(Enter as many time values as required	
31-40	E10.0	using as many columns and cards as	
41-50	E10.0	necessary. Final time should be equal	
51-60	E10.0	to or greater than final calculation	
61-70	E10.0	time)	
71-80	E10.0		

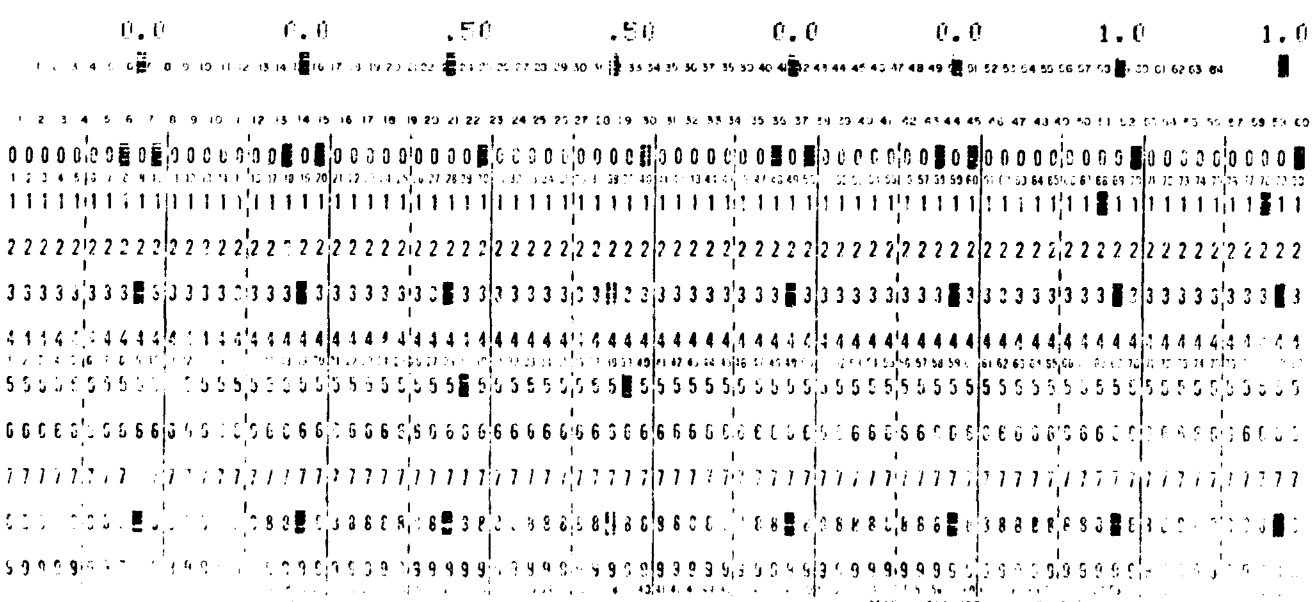
EXAMPLE CARD



CARD NUMBER 4

COLUMN	FORMAT	DATA	DIMENSIONS
1-10	E10.0	Initial valve position or opening T = 0.0	IN or -
11-20	E10.0	(Enter valve positions or portions of	
21-30	E10.0	rated flow [at full rated pressure drop]	
31-40	E10.0	that correspond to the times given on the	
41-50	E10.0	previous card(s)).	
51-60	E10.0		
61-70	E10.0		
71-80	E10.0		

EXAMPLE CARD



6.21 SUBROUTINE VALV21

VALV21 simulates a simple two way valve with a valve opening/time history input. It can be used to simulate directly operated solenoid valves or balanced poppet valves, but not valves which have displacement characteristics such as two stage or unbalanced poppet valves.

Input data can be in the form of slot width, coefficient of discharge and slot opening, or rated flow, rated pressure drop and fraction of rated flow.

A typical valve which can be modeled by VALV21 is shown in Figure 6.21-1.

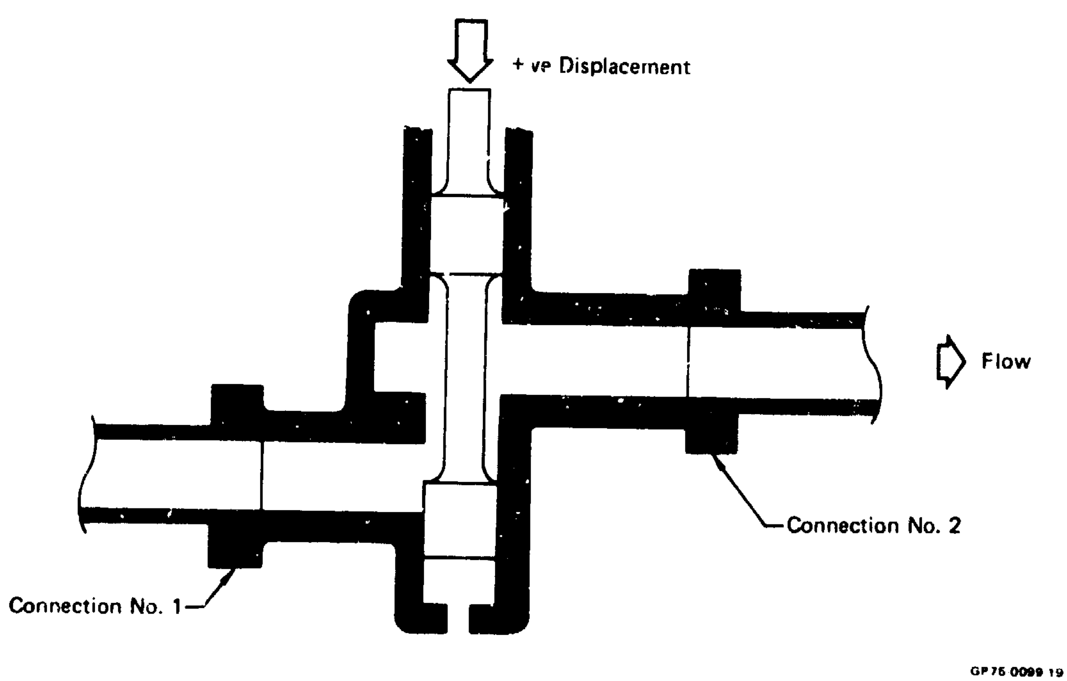


FIGURE 6.21-1
TYPE NO. 21 TWO-WAY VALVE

6.21.1 MATH MODEL

VALV21 is modeled as an orifice of variable area. As such, it follows the basic equation for flow through an orifice.

$$Q = AREA*CD * (2*(P1-P2)/RHO)^{1/2} \tag{1}$$

where

AREA = area of the valve orifice (in^2)

CD = discharge coefficient of the valve

P1 = inlet pressure PSI

P2 = outlet pressure PSI

RHO = fluid density ($lb*sec^2/in^4$)

Using a positive flow convention from P1 to P2, the line equations for the component are:

$$P1 = C1 - Z1 * Q1 \tag{2}$$

$$P2 = C2 + Z2*Q2 \tag{3}$$

where

C1 =	connection #1 line characteristic	(PSI)
C2 =	connection #2 line characteristic	(PSI)
Z1 =	connection #1 line impedance	(PSI/CIS)
Z2 =	connection #2 line impedance	(PSI/CIS)

When assuming no flow loss:

$$Q = Q1 = Q2 \tag{4}$$

Substitution of equations (2) and (3) into equation (1) gives

$$Q = AREA*CD*(2*C1-Z1*Q-C2-Z2*Q)/RHO)^{1/2} \tag{5}$$

Then squaring both sides of equation (5) gives

$$Q^2 = AREA^2*CD^2*2/RHO*[(C1-C2)-Q(Z1+Z2)] \tag{6}$$

By defining $VFAC = AREA^2*CD^2*2/RHO$

$CDIFF = C1-C2$

$ZTOT = Z1+Z2$

Equation (6) will reduce to:

$$Q^2+VFAC*ZTOT*Q - VFAC*CDIFF = 0 \tag{7}$$

When simplified by the definition of:

$VB = VFAC*ZTOT$

$VC = CDIFF*VFAC$

Equation (7) becomes

$$Q^2+VB*Q-VC = 0$$

which is a quadratic equation with the solution

$$Q = (-VB \pm(VB^2+4VC)^{1/2})/2$$

6.21.2 CAVITATION MODEL

The cavitation model of VALV21 is the same as that described in subroutine REST41 (Section 6.41.2).

6.21.3 ASSUMPTIONS

The math model assumes a square law characteristic and a constant discharge coefficient for the entire flow range. In practice, neither of these assumptions is correct. Most valves have inlet and outlet passages which cause the flow characteristics to vary from the square law relationship. The discharge coefficients can vary with area change and flow rate. A subroutine that included these considerations would be very complex and involve input data not readily available to the average user.

In view of this, the model provided by VALV21 should be adequate for most simulations.

6.21.4 COMPUTATION METHOD

1000 SECTION

A check is made on which type of data is being used. If the rated flow/pressure drop data is used, a density correction is made between the simulation fluid and valve rated fluid. A valve characteristic is computed along with the steady state calculation constants.

If the valve slot dimension input is used, the valve characteristic and steady state constants are calculated directly from the fluid density and valve area at time T = 0.

1500 SECTION

The steady state constants from the previous section are used in the Q^2 leg calculations, and the pressure at the outlet connection is calculated and stored.

2000 SECTION

The DT variables, which will be used to track the cavitation bubbles, are zeroed and control is returned to the main program.

3000 SECTION

A call to INTERP is made and the new valve position determined using a first degree interpolation. A check is made of the DT variables to determine if the component is cavitated. If it is cavitated, the appropriate boundary conditions are applied and the orifice equation is used to compute the flow. Upstream and downstream pressures and flows are calculated and the cavitation bubble updated before control is returned to the main program. If the valve is not cavitated, the normal line and orifice equations are used to calculate the flow and pressures.

A check is made to see if the valve has cavitated during the time step under calculation. If it has, flows and pressures are recalculated according to the cavitation model. If it has not cavitated, control is returned to the main program.

6.21.5 LIMITATIONS

The computation is limited to a linear valve area versus position relationship. However, a position/time history that approximates the actual valve area/position relationship can be input. A straight line interpolation is used to avoid unintended valve motion which could result with higher degree interpolation methods. The limitations can be minimized by using smaller intervals in the time history table.

If the valve slot width and discharge coefficient are input as 1.0, the valve position table becomes a table of the product of valve area and discharge coefficient versus time. This can, in part, rectify the assumption of a constant discharge coefficient, but it can not correct for the variation with flow rate.

6.21.6 Variable Names

NAME	DESCRIPTION	DIMENSIONS
D(3)	Valve Characteristic	CIS/PSI*SEC
DENS	Rated Fluid Density	$LB*SEC^2/IN^4$
DENSC	Density Correction Factor	--
DIFF	Characteristic Pressure Difference	PSI
DT(1)	Connection #1 Cavitation Tracker	CIS
DT(2)	Connection #2 Cavitation Tracker	CIS
IFLUID	Rated Fluid Code	--
P1	Connection #1 Cavitation Pressure	PSI
P2	Connection #2 Cavitation Pressure	PSI
QX	Normal Orifice Flow	CIS
QV	Cavitated Orifice Flow	CIS
RTEMP	Rated Temperature	DEGREES F
VALY	Valve Position	IN
VFAC	Valve Orifice Characteristic	CIS^2/PSI
VYMAG	Valve Opening	IN
ZT	Cavitation Line Impedance	PSI/CIS

6.21.7 Subroutine Listing

```
      SUBROUTINE VALV21 (D,DT,DD,L)
C *** REVISED DECEMBER 1979 ***
C *** THIS SUBROUTINE INCORPORATES A CAVITATION MODEL ON THE UPSTREAM
C *** AND DOWNSTREAM SIDES OF THE VALVE. IT ALSO INCLUDES THE OPTION OF
C *** SPECIFYING VALVE CHARACTERISTICS BY SLOT DIMENSION AND
C *** COEFFICIENT OF DISCHARGE OR BY RATED PRESSURE DROP AND RATED FLOW
      DOUBLE PRECISION DD
      COMMON NTELPL,NTOLPL,IPT,IPOINT,NPTS,INEL,KNEL,NTOPL,NLPLT(61,3),
     1 PQLEG(90,12),LCS(90,10),ILEG(1400),PN(90),QN(90)
      DIMENSION D(24),DT(2),DD(1),L(5)
      COMMON/SUB/PARM(150,9),PM(1500),QM(1500),P(300),Q(300),C(300)
     1,Z(300),RHO(20),S2ORHO(20),VISC(20),BULK(20),TEMP(20),PVAP(20)
     2,ATPRES,T,DELT,TFINAL,PLTDEL,PI,TITLE(20),LEGN,ICON
     3,KTEMP(99),LSTART(150),NLPT(150),LTYPE(99),NC(99),INX,INZ
     4,INV,ISTEP,NLINE,NEL,IND,IENTR,MNLINE,MNEL,MNLEG,MNNODE,MNPLOT
     5,MNLPTS,MDS
      IF (IENTR) 1000,2000,3000
 1000 CONTINUE
      IF(INEL.NE.0) GO TO 1500
      L(5)=9
      L(6)=9+(L(11)+7)/8*8
      L(7)=L(11)
      D(4)=(Z(L(1))+Z(L(2)))/2.
      D(5)=1.0/D(4)**2
      VALY=D(L(6))
      IF(L(3).NE.0) GO TO 1001
C *** CALCULATIONS FOR ORIFICE DIMENSION-COEFFICIENT OF DISCHARGE INPUT
      D(3)=(D(1)*D(2))**2*2/RHO(KTEMP(IND))
      DT(1)=RHO(KTEMP(IND))/(2.*(D(1)*VALY*D(2))**2)
      DT(2)=0.0
      IF(VALY.EQ.0.0) GO TO 1100
      RETURN
C *** CALCULATIONS FOR RATED CONDITION INPUT
 1001 IF(L(4).EQ.0) GO TO 1003
C *** DENSITY CALCULATION FOR RATED CONDITIONS
      IFLUID=L(4)
      RTEMP=D(3)
      IF(IFLUID-2)1004,1005,1006
C *** DENSITY CALCULATION OF MIL-H-5606B AT RATED TEMPERATURE
 1004 DENS=56.0147-1.82353E-2*RTEMP
      GO TO 1007
C *** DENSITY CALCULATION OF MIL-H-83282 AT RATED TEMPERATURE
 1005 DENS=55.0706-2.35294E-2*RTEMP
      GO TO 1007
C *** DENSITY CALCULATION OF SKYDROL 500B AT RATED TEMPERATURE
 1006 DENS=67.0221-2.73529E-2*RTEMP
      GO TO 1007
C *** DEFAULT DENSITY
 1003 DENS=54.19117
C *** CHANGE TO CORRECT DENSITY UNITS
```

6.21.7 (Continued)

```
 1007 DENS=DENS/(12.**4.*32.174)
C *** DENSITY CORRECTION TO COMPONENT CONDITIONS
      DENSC=DENS/RHO(KTEMP(IND))
      DT(1)=D(2)/((D(1)*VALY)**2.*DENSC)
      D(3)=D(1)**2.*DENSC/D(2)
      DT(2)=0.0
      IF(VALY.EQ.0.0) GO TO 1100
      RETURN
 1100 DT(1)=0.0
      DT(2)=10.E5
      RETURN
C
C**** STEADY STATE SECTION ****
C
 1500 PQLEG(INEL,8)=PQLEG(INEL,8)+DT(1)
      QX=PQLEG(INEL,1)
      PQLEG(INEL,6)=PQLEG(INEL,6)+DT(2)
      PQLEG(INEL,11)=PQLEG(INEL,11)-(PQLEG(INEL,2)*QX*(DT(2)+QX*DT(1)))
      RETURN
 2000 CONTINUE
      DT(1)=0.0
      DT(2)=0.0
      RETURN
 3000 CONTINUE
      L1=L(1)
      L2=L(2)
      CALL INTERP (T,D(L(5)),D(L(6)),10,L(7),VALY,IOD)
      VYMAG= ABS(VALY)
      DIFF=C(L1)-C(L2)
      IF(DT(1)+DT(2).GT.0.0)GO TO 3300
      IF(VYMAG.LT.0.00001) GO TO 3100
      VFAC=D(3)*VALY**2
      QX=SIGN(VFAC*D(4)*(SQRT(1+ABS(DIFF)/VFAC*D(5))-1),DIFF)
      P(L1)=C(L1)-Z(L1)*QX
      P(L2)=C(L2)-Z(L2)*(-QX)
      Q(L1)=QX
      Q(L2)=-QX
      GO TO 3200
 3100 CONTINUE
      P(L1)=C(L1)
      P(L2)=C(L2)
      Q(L1)=0.0
      Q(L2)=0.0
 3200 IF(P(L1).GE.PVAP(KTEMP(IND)).AND.P(L2).GE.PVAP(KTEMP(IND)))RETURN
      IF(P(L1).LT.PVAP(KTEMP(IND)).AND.P(L2).LT.PVAP(KTEMP(IND)))
     + GO TO 3230
      IF(P(L1).LT.PVAP(KTEMP(IND)))GO TO 3220
 3210 P2=PVAP(KTEMP(IND))
      ZT=Z(L1)
      P1=C(L1)
```

6.21.7 (Continued)

```
      IF(C(L1).GT.PVAP(KTEMP(IND)))GO TO 3500
      P1=PVAP(KTEMP(IND))
      GO TO 3600
 3220 P1=PVAP(KTEMP(IND))
      ZT=Z(L2)
      P2=C(L2)
      IF(C(L2).GT.PVAP(KTEMP(IND)))GO TO 3500
      P2=PVAP(KTEMP(IND))
      GO TO 3600
 3230 IF(P(L1)-P(L2))3210,3400,3220
 3300 IF(DT(1).GT.0.0.AND.DT(2).GT.0.0)GO TO 3400
      IF(DT(1).GT.0.0)GO TO 3220
      GO TO 3210
 3400 P1=PVAP(KTEMP(IND))
      P2=PVAP(KTEMP(IND))
      GO TO 3600
 3500 IF(VYMAG.LT..00001)GO TO 3600
      VFAC=D(3)*VALY**2
      QV=VFAC*ZT/2.0*(SQRT(1.0+4.0*ABS(P1-P2)/(VFAC*ZT**2))-1.0)
      QV=SIGN(QV,P1-P2)
      IF(P1.GT.PVAP(KTEMP(IND)))GO TO 3700
      GO TO 3800
 3600 QV=0.0
      GO TO 3750
 3700 P(L1)=C(L1)-QV*Z(L1)
      Q(L1)=QV
 3750 Q(L2)=(C(L2)-P2)/Z(L2)
      P(L2)=P2
      DT(2)=DT(2)-QV-Q(L2)
      IF(QV.EQ.0.0)GO TO 3850
      RETURN
 3800 P(L2)=C(L2)-QV*Z(L2)
      Q(L2)=QV
 3850 Q(L1)=(C(L1)-P1)/Z(L1)
      P(L1)=P1
      DT(1)=DT(1)+QV-Q(L1)
      RETURN
      END
```

6.31 TYPE #31 CHECK VALVE

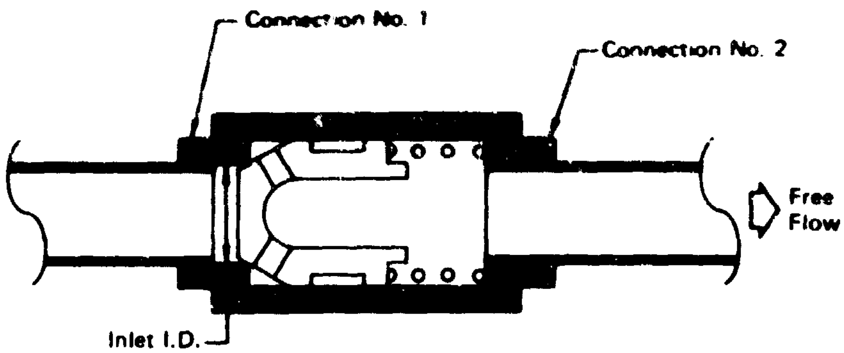


FIGURE 6.31-1
TYPE NO. 31 CHECK VALVE

The TYPE #31 component is a model of a simple poppet type check valve. It models the poppet dynamics and flow characteristics of the valve and can have any damping factor the user wants to input.

The input data is quite detailed for such a simple device. Modeling of dynamic response characteristics requires detailed design parameters. If such information is unknown, the component data handbook can be used to estimate typical values for the input data.

The component does not have any cavitation provisions, so it is possible that negative pressures will result when it is used.

CARD NUMBER 1

COLUMN	FORMAT	DATA
1-5	I5	Component Number
6-10	I5	Type Number = 31
11-15	I5	Number of Real Data Cards = 1
16-20	I5	Line Number (with sign) attached to Connection 1
21-25	I5	Line Number (with sign) attached to Connection 2
26-30	I5	Connection Order (0 or +1 - conventional)
31-35	I5	(-1 - backwards)
36-40	I5	
41-45	I5	
46-50	I5	
51-55	I5	
56-60	I5	
61-65	I5	
66-70	I5	
71-75	I5	
76-80	I5	Temperature/Pressure Code (See Page 4.0-2)

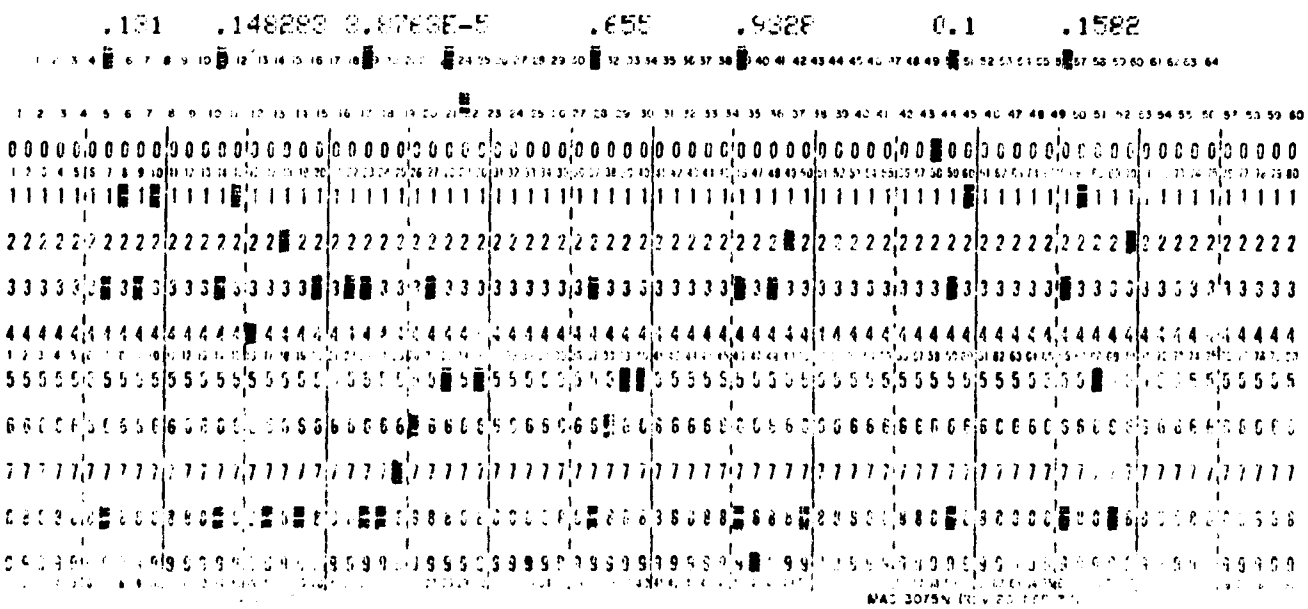
EXAMPLE CARD

9 31 1 +7 -8 0

CARD NUMBER 2

COLUMN	FORMAT	DATA	DIMENSIONS
1-10	E10.0	Poppet Area	IN^2
11-20	E10.0	Full Flow Area	IN^2
21-30	E10.0	Poppet Mass	$LB*SEC^2/IN$
31-40	E10.0	Spring Preload	LB
41-50	E10.0	Spring Constant	LB/IN
51-60	E10.0	Damping Factor	LB*SEC/IN
61-70	E10.0	Maximum Poppet Displacement	IN
71-80	E10.0		

EXAMPLE CARD

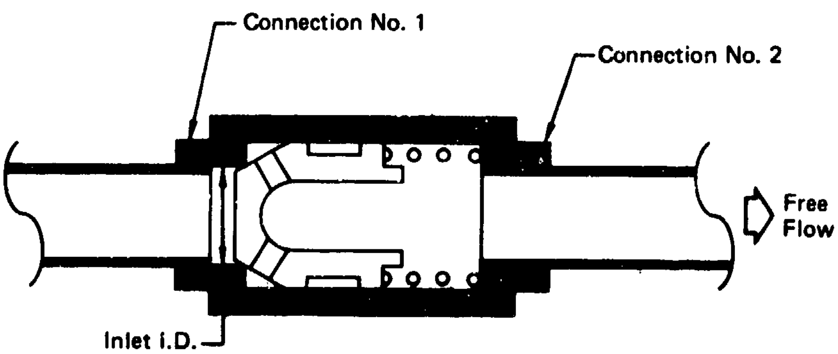


6.31 SUBROUTINE CVAL31

Subroutine CVAL31 models a simple poppet type check valve. The check valve may be damped or undamped. Figure 6.31-1 shows an undamped check valve of the type generally in use in the aircraft industry today. CVAL31 models the poppet dynamics and flow characteristics of such a valve when subjected to transient pressures.

Subroutine CVAL31 does not have a cavitation model.

The input data for the model is quite detailed. Accurate dynamic response requires detailed design parameters. If such information is unknown, the component data handbook can be used to estimate typical values for the input data.



OP74-0773-8

FIGURE 6.31-1
TYPE NO. 31 CHECK VALVE

6.31.1 MATH MODEL

The check valve is modeled as an orifice of variable area. The area at any given time is determined by the position of the force balanced poppet. CVAL31 uses the poppet position of the last time step to calculate the flow and pressure drop through the check valve. With the upstream and downstream pressures known, a force balance is performed on the poppet to determine its new position. The new position is used in the flow calculation and an iteration process between the flow calculation and poppet position occurs until a correct poppet position is reached.

Figure 6.31-2 shows the forces acting on the check valve poppet.

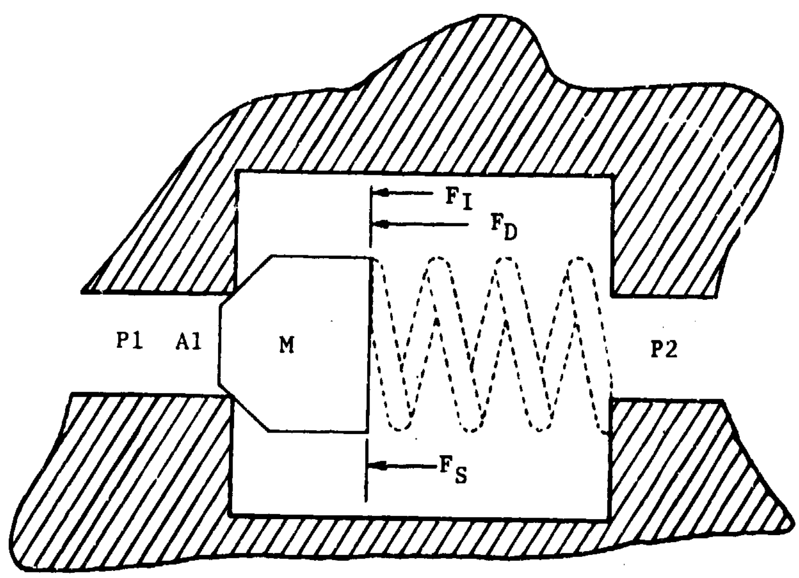


Figure 6.31-2

Defining x = o as the poppet displacement when fully seated, the force balance on the poppet is

$$(P1-P2) * A1 = F_I + F_D + F_S \qquad (1)$$

Where $P1$ = inlet pressure (PSI)

$P2$ = outlet pressure (PSI)

F_I = poppet inertia force (LB)

F_D = poppet damping force (LB)

F_S = spring force (LB)

$A1$ = POPPET AREA (IN^2)

By further defining

$$F_S = P_L + K_S * X_N$$

$$FD = K_D * V_N$$

$$F_I = M * a$$

Where P_L = spring preload (LB)

K_S = spring constant (LB/IN)

X_N = present poppet displacement (IN)

K_D = damping factor (LB*SEC/IN)

V_N = present poppet velocity (IN/SEC)

M = poppet mass ($LB*SEC^2/IN$)

a = poppet acceleration (IN/SEC^2)

And using the physical definitions for velocity and acceleration

$V_N = (Xn - Xo)/\Delta t$ (IN/SEC)

$Q = (Vn - Vo)/\Delta t$ (IN/SEC^2)

Where Xo = previous poppet position (IN)

Vo = previous poppet velocity (IN/SEC)

Δt = time increment (SEC)

Making all of the substitutions, the force balance (Equation 1) becomes

$$(P1-P2)*A1 = M* (Xn-Xo-Vo*\Delta t)/\Delta t^2 + K_D* (Xn-Xo)/\Delta t + P_L + K_S * X_N \qquad (2)$$

Using Equation (2), the present poppet displacement can be solved for as

$$X_N = [(P1-P2)/A1+Xo* (\frac{M}{\Delta t^2} + \frac{KD}{\Delta t}) + \frac{Vo * M}{\Delta t} - P_L] / \left[\frac{M}{\Delta t^2} + \frac{KD}{\Delta t} + KS\right] \qquad (3)$$

Using the line equations

$$P1 = C1 - Z1*Q1 \qquad (4)$$

$$P2 = C2 * Z2*Q2 \qquad (5)$$

Where a positive flow convention from P1 to P2 is assumed and,

C1 = connection #1 line characteristic (PSI)

C2 = connection #2 line characteristic (PSI)

Z1 = connection #1 line impedance (PSI/CIS)

Z2 = connection #2 line impedance (PSI/CIS)

Assuming conservation of flow.

$$Q = Q1 = Q2$$

The pressure differential necessary for Equation (3) can be written as

$$P1-P2 = C1-C2 - Q* (Z1 + Z2) \qquad (6)$$

Using the equation for flow through an orifice

$$Q = AV*CV* (2* (P1-P2)/RHO)^{1/2}$$

Where Q = flow (CIS)

AV = orifice area (IN^2)

CV = discharge coefficient

RHO = fluid density ($LB*SEC^2/IN^4$)

with Equation (2) gives the quadratic equation

$$Q^2 + Q*R^2*ZTOT - R^2*CDIFF = 0 \qquad (7)$$

Where R = $AV*CV*2/\sqrt{RHo}$ ($IN^4/(SEC*LB^{1/2})$)

ZTOT = Z1+Z2 ($LB*SEC/IN^5$)

CDIFF = C1-C2 (LB/IN^2)

The solution to Equation (7) is

$$Q = (-ZTOT*R^2 \pm \sqrt{R^4*ZTOT^2 - 4*R^2*CDIFF})/2.0 \qquad (8)$$

With the flow from Equation (8), Equation (6) can be used to determine the pressure drop necessary for the poppet force balance (Equation 3). Assuming the linear position vs. area rule of the poppet, the variable R in Equation (7) can be updated each iteration by the formula

$$R = R*X_N/Xmax \qquad (9)$$

Where Xn = poppet position (calculated by Equation 3)

Xmax = maximum poppet displacement

In subroutine CVAL31, an iteration using Equations 8, 6, 3 and 9 is used to converge to the correct poppet position for a given time step.

6.31.2 LIMITATIONS

The model does not account for displacement flow due to poppet motion, for the variations in orifice characteristic with poppet position, for secondary pressure drops due to other flow restrictions, or for flow forces on the poppet. All of these factors are highly dependent on individual valve design and to include them would result in a model too complex and specific to be of any general use.

As a result of these limitations, the dynamic simulation of the check valve may not be very accurate, especially when excited by very fast transients or when in a simulation where a large time step is used. In simulations where the poppet will be either fully open or fully closed for the majority of the time, the above limitations are acceptable and the model should give good results.

6.31.3 COMPUTATIONS

1000 SECTION - Valve characteristics necessary for the steady state portion of the program are calculated and stored in the component's DT array. Control is then returned to the main program.

1500 SECTION - A series of checks are made to determine the orientation of the component with respect to leg and line assemblies. If the flow direction given by the main program is opposite the free flow direction of the check valve, the leg constant for the flow term is set equal to 10^6 PSI/CIS, the upstream pressure is calculated, and control is returned to the program.

If the flow is in the free flow direction of the valve and is greater than the flow that would result through the fully open valve at the pressure drop necessary to fully open the poppet, the lower flow limit of the leg is set equal to the fully open flow. The orifice constant for a fully open poppet is used for the Q^2 leg calculation and the upstream pressure is calculated and stored.

If the flow is in the free flow direction but is not sufficient enough to hold the poppet open, the upstream pressure is recalculated using the formula.

$$P_{up} = P_{up} - Pc * Qs - Qa * Qs * S$$

Where P_{up} = upstream pressure (PSI)

Pc = cracking pressure = spring preload/poppet area (PSI)

Qs = flow sign (+1 or -1)

Qa = absolute value of flow guess (CIS)

S = slope of intermediate position pressure drop (PSI/CIS)

= (spring const. * max disp) / (flow through fully open poppet at ΔP for full opening)

Poppet position is then calculated and stored using the formula

$$\text{Poppet position} = (S*Qa+Pc)/P_{open} * \text{MAX DISP}$$

Where P_{open} = pressure drop necessary to fully open poppet

A check is then made to determine if the upper flow limit is greater than the minimum flow through the fully open poppet. If it is, the upper flow limit is set equal to the minimum flow. The cracking pressure is multiplied by the flow sign then subtracted from the constant pressure drop leg term and the slope of the intermediate position pressure drop/flow line (S) is added to the Q leg term before control is returned.

2000 SECTION - The DT variables are initialized for the transient section.

3000 SECTION - A check is made to determine the last time step's poppet position. If the poppet was closed, flow is set equal to zero. If the poppet was open, flow is calculated using Equations (8) and (9) of Section 6.31.1. With this flow, Equation (6) of Section 6.31.1 is used to determine the pressure drop necessary for calculation of the new poppet position (Equation 3). Limits are placed on this new poppet position to keep it within the operating displacement of the check valve poppet. The new position is compared to the last calculation of the poppet position and if the difference is less than $\pm$ .001 inches, new poppet velocity and displacement are calculated and stored, then the line pressures and flows determined before control is returned.

If the new poppet position is more than .001 inches different than the last calcualtion of poppet position, they are averaged and the flow is again calculated using this averaged poppet position. An iteration using the flow, pressure drop, and poppet position equations is made until the difference in poppet positions between one iteration and the next is less than .001 inches. (The iteration is limited to 10) Once the poppet position is determined, it is stored as the time step's calculation of poppet position and used in the calculation of poppet velocity. The flow that was calculated in the last iteration is used for the line flow and line pressure calculations, then control is returned to the main program.

6.31.4 ASSUMPTIONS

The assumptions of no flow forces on the poppet and a linear area vs displacement relationship can be sources of inaccuracy. Also, the assumption of a square law (orifice) flow characteristic is not borne out by manufacturer's test data. All of these factors can be highly design dependent, and hence for a simple model provide some justification for the assumptions.

6.31.5 LIMITATIONS

The main limitation of subroutine CVAL31 is the lack of a cavitation simulation. This can result in negative pressures being calculated when pressures drop below the fluid vapor pressure.

6.31.6 Variable Names

NAME	DESCRIPTION	DIMENSIONS
CDIFF	Characteristic Pressure Difference	PSI
D(A1)	Poppet Area	IN^2
D(A2)	Full Flow Orifice Area	IN^2
D(DAMP)	Poppet Damping Factor	LB*SEC/IN
D(PL)	Spring Preload	LB
D(XMAX)	Maximum Poppet Displacement	IN
DP	Pressure Drop Across Poppet	PSI
DT(CRV)	Flow Characteristic of Poppet	CIS/ PSI*IN
DT(CXO)	Poppet Inertia/Damping Characteristic	LB/IN
DT(K1)	Poppet Spring/Mass Characteristic	LB/IN
DT(VO)	Poppet Velocity	IN/SEC
DT(X)	Poppet Position	IN
ITER	Iteration Counter	--
Q1	Component Flow	CIS
QA	Leg Flow	CIS
QS	Flow Sign	--
RV	Flow Characteristic at Given Poppet Displacement	CIS/ PSI
XN	New Poppet Position	IN
XP	Predicted Poppet Position	IN
ZT	Total Line Characteristics	PSI

6.31.7 Subroutine Listing

```
      SUBROUTINE CVAL31 (D,DT,DD,L)
C *** REVISED DECEMBER 1979 ***
C *** THIS IS A CHECK/RELIEF VALVE MODEL THAT INCLUDES THE POPPET
C *** DYNAMICS.
      DIMENSION D(1),DT(1),DD(1),L(1)
      DOUBLE PRECISION DD
      COMMON NTELPL,NTOLPL,IPT,IPOINT,NPTS,INEL,KNEL,NTOPL,NLPLT(61,3),
     1 PQLEG(90,12),LCS(90,10),ILEG(1400),PN(90),QN(90)
      COMMON/SUB/PARM(150,9),PM(1500),QM(1500),P(300),Q(300),C(300)
     1,Z(300),RHO(20),S2ORHO(20),VISC(20),BULK(20),TEMP(20),PVAP(20)
     2,ATPRES,T,DELT,TFINAL,PLTDEL,PI,TITLE(20),LEGN,ICON
     3,KTEMP(99),LSTART(150),NLPT(150),LTYPE(99),NC(99),INX,INZ
     4,INV,ISTEP,NLINE,NEL,IND,IENTR,MNLINE,MNEL,MNLEG,MNNODE,MNPLOT
     5,MNLPTS,MDS
      INTEGER A1,A2,CRV,CXO,DAMP,PL,VO,X,XMAX
C *** FOR THE STEADY STATE SECTION THE DT VARIABLES HAVE THE
C *** FOLLOWING DEFINITIONS.
C DT(1)= CRACKING PRESSURE
C DT(2)= FULL OPEN PRESSURE
C DT(3)= FULL OPEN FLOW
C DT(5)= SLOPE OF INTERMEDIATE POSITION PRESSURE DROP
C *** THE DT VARIABLES ARE REDEFINED IN THE 2000 AND 3000 SECTIONS.
C *** D VARIABLES ***
      DATA A1/1/,A2/2/,M/3/,PL/4/,KS/5/,DAMP/6/,XMAX/7/
C *** DT VARIABLES ***
      DATA X/1/,VO/2/,CXO/3/,CRV/4/,K1/5/
      IF(IENTR) 1000,2000,3000
 1000 CONTINUE
      IF(INEL.NE.0) GO TO 1500
      DT(CRV)=(D(A2)/D(XMAX))*.65*S2ORHO(KTEMP(IND))
      DT(1)=D(PL)/D(A1)
      DT(2)=D(KS)*D(XMAX)/D(A1)+DT(1)
      DT(3)=D(A2)*.65*S2ORHO(KTEMP(IND))*SQRT(DT(2))
      DT(5)=(DT(2)-DT(1))/DT(3)
      RETURN
C      SECTION FOR STEADY STATE CALCULATION
 1500 CONTINUE
      IF(KNEL.NE.1)RETURN
      QA=PQLEG(INEL,1)
      QS=PQLEG(INEL,2)
      LCS(INEL,6)=QS
      IF(L(3).EQ.1.) GO TO 1600
C     THE VALVE IS CONNECTED CONVENTIONALY
      IF(L(3).NE.0)GO TO 1900
      IF(QS.EQ.1.) GO TO 1700
      GO TO 1650
C     THE VALVE IS CONNECTED BACKWARDS
 1600 IF(L(3).NE.1)GO TO 1900
      IF(QS.LT.1.0) GO TO 1700
C     THE VALVE IS CLOSED
```

```
 1650 PQLEG(INEL,6)=1.0E6+PQLEG(INEL,6)
      PQLEG(INEL,11)=PQLEG(INEL,11)-QA*QS*1.0E6
      DT(6)=0.0
      RETURN
 1700 IF (QA.LE.DT(3)) GO TO 1800
C     THE VALVE IS FULLY OPEN
      IF(PQLEG(INEL,3).LT.DT(3)) PQLEG(INEL,3)=DT(3)
      DT(6)=D(XMAX)
      PQLEG(INEL,8)=PQLEG(INEL,8)+1./(DT(6)*DT(CRV))**2
      PQLEG(INEL,11)=PQLEG(INEL,11)-QS*QA**2/(DT(6)*DT(CRV))**2
      RETURN
C     THE FLOW IS LESS THAN THE 'FULL OPEN FLOW'
 1800 PQLEG(INEL,11)=PQLEG(INEL,11)-DT(1)*QS
     1 -QA*QS*DT(5)
      DT(6)=((DT(5)*QA+DT(1))/DT(2))*D(XMAX)
      IF(PQLEG(INEL,4).GT.DT(3)) PQLEG(INEL,4)=DT(3)
      PQLEG(INEL,5)=PQLEG(INEL,5)-DT(1)*QS
      PQLEG(INEL,6)=PQLEG(INEL,6)+DT(5)
      RETURN
C*****ILLEGAL INPUT DATA TERMINATES PROGRAM*****
 1900 WRITE(6,998)
  998 FORMAT(10X,33HPROGRAM STOP IN SUBROUTINE CVAL31)
      STOP 1531
 2000 CONTINUE
      DT(CXO)=D(M)/DELT**2+D(DAMP)/DELT
      DT(K1)=DT(CXO)+D(KS)
      DT(X)=DT(6)
      DT(VO)=0.0
      RETURN
 3000 CONTINUE
      L1=L(1)
      L2=L(2)
      ITER=0
      ZT=Z(L1)+Z(L2)
      CDIFF=C(L1)-C(L2)
      XO=DT(X)
      XP=DT(X)
 3010 RV=XP*DT(CRV)
      IF(RV.GT.0.0) GO TO 3020
      Q1=0.0
      GO TO 3030
 3020 Q1=SIGN((RV**2*ZT/2.)*(-1.0+SQRT(1.0+4.0*ABS(CDIFF)
     +/(RV**2*ZT**2))),CDIFF)
 3030 DP=CDIFF-Q1*ZT
      XN=(DP*D(A1)+XO*DT(CXO)+DT(VO)*D(M)/DELT-D(PL))/DT(K1)
      IF(XN.LT.0.0005) XN=0.0
      IF(XN.GT.D(XMAX)) XN=D(XMAX)
      IF(ABS(XN-XP).LT..001) GO TO 3050
      XP=(XN+XP)/2.0
      IF(XP.LT.0.0005) XP=0.0
```

6.31.7 (Continued)

```
      IF(XP.GT.D(XMAX)) XP=D(XMAX)
      IF(ITER.GE.10) GO TO 3050
      ITER=ITER+1
      GO TO 3010
3050  IF(XN.EQ.0.0)XP=0.0
      DT(VO)=(XP-DT(X))/DELT
      WRITE(6,4000)DP,CDIFF,XN,XP,XO,Q1,ITER
4000  FORMAT(2X,6HCVAL31,2X,6E12.4,2X,1I3)
      IF(XP.EQ.0.0) DT(VO)=C.0
      DT(X)=XP
      IF(XP.LE.0.0) Q1=0.0
      Q(L1)=Q1
      Q(L2)=-Q1
      P(L1)=C(L1)-Q(L1)*Z(L1)
      P(L2)=C(L2)-Q(L2)*Z(L2)
      RETURN
      END
```

6.32 TYPE #32 PRIORITY VALVE

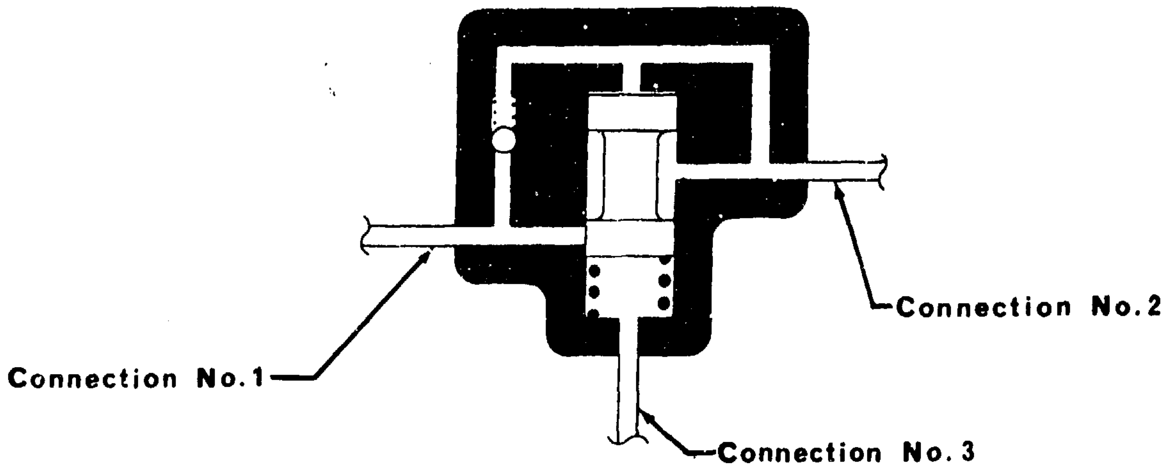


FIGURE 6.32-1
TYPE NO. 32 PRIORITY VALVE

The Type #32 priority valve is modeled as a parallel check valve/relief valve combination. As shown in Figure 6.32-1, the priority valve allows free flow from the outlet (Connection #1) to the inlet (Connection #2) through the check valve, and permits reverse flow when the pressure at Connection #2 is sufficient to open the relief valve.

In practice, devices such as this give flow priority to the subsystems upstream of them by shutting off flow when their inlet pressure drops below a specified level.

The priority valve model is an idealized model. As such, it is instantaneous in its reactions and does not have the spring/mass/damping effects. It should give accurate results in simulations where it is not excited by fast transient pressures around its shutoff value.

6.32 SUBROUTINE CREL32

Subroutine CREL32 models a hydraulic priority valve with free reverse flow. As diagramed in Figure 6.32-1, the subroutine simulates the component as a parallel relief valve/check valve combination. The check valve allows flow in the free flow (reverse) direction and the relief valve modulates flow in the normal direction (Connection #2 to Connection #1).

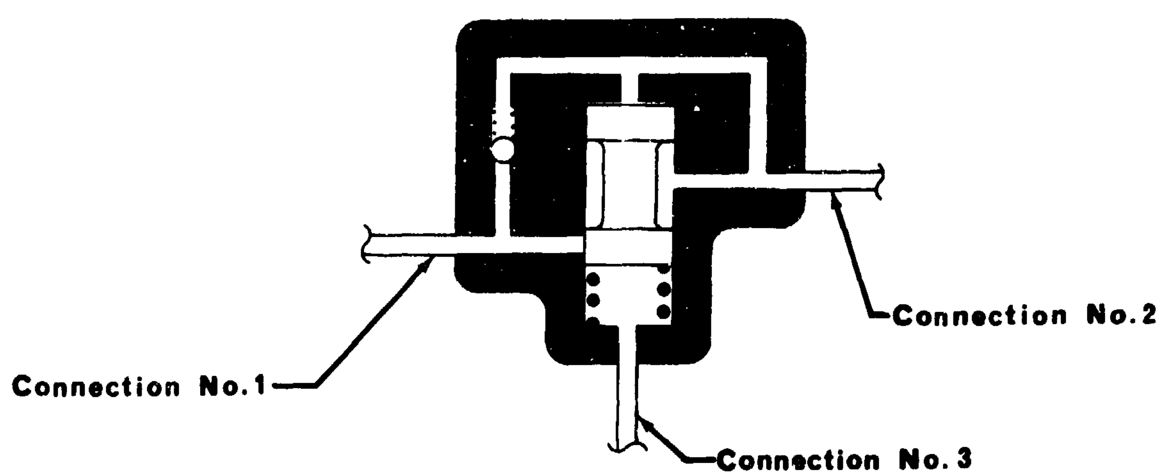


FIGURE 6.32-1
TYPE NO. 32 PRIORITY VALVE

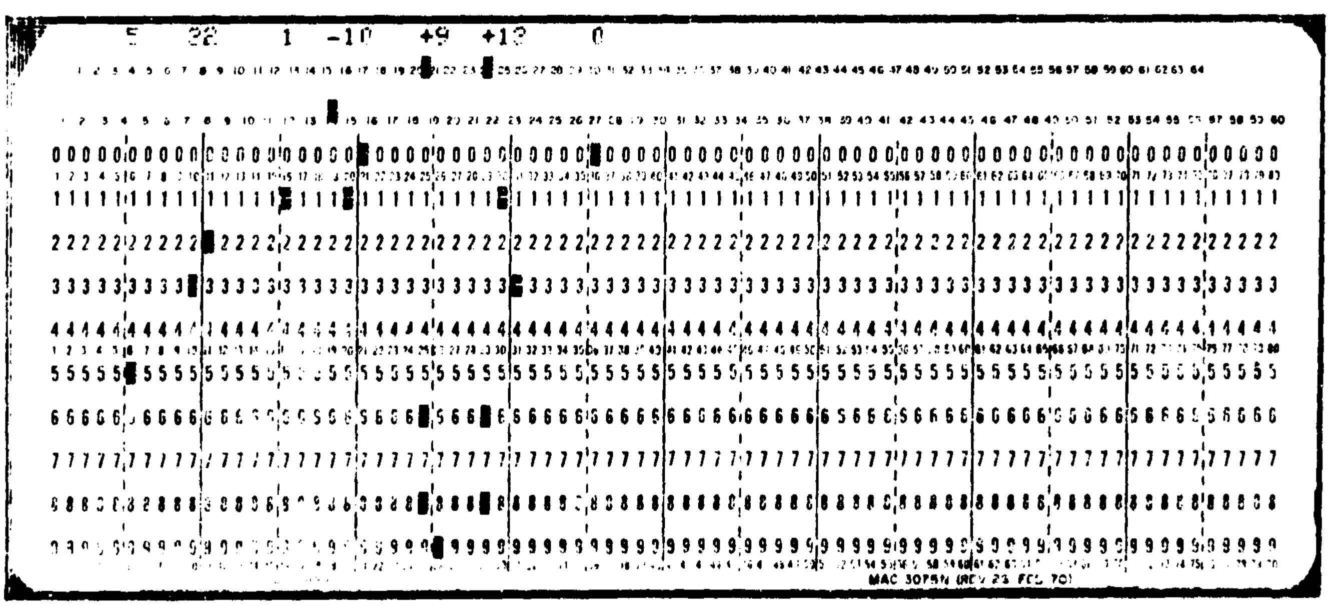
Connection #3 in an actual priority valve is used for relief valve reference pressure and as a return path for any internal seal leakage. In this model, Connection #3 serves no direct purpose. It is recommended that no node be placed at a Type 32 priority valve and that the steady state leg attached to the valve be assembled directly through it (Connection #1 to Connection #2 or vice versa). Even though Connection #3 sees no use, the line number attached to it must be input on the first card of the component. It is easiest to handle this line by coding it as a dead ended line and assembling it as a zero flow leg originating at the branch where it ties into the return system.

During the steady state calculations, the program assumes the priority valve is open in both directions when the system is pressurized and closed when the system is depressurized.

CARD NUMBER 1

COLUMN	FORMAT	DATA
1-5	I5	Component Number
6-10	I5	Type Number = 32
11-15	I5	Number of Real Data Cards = 1
16-20	I5	Line Number (with sign) attached to Connection 1
21-25	I5	Line Number (with sign) attached to Connection 2
26-30	I5	Line Number (with sign) attached to Connection 3
31-35	I5	Operation Code (+1) - Both Valves Closed
36-40	I5	(0) - Relief Valve Opened
41-45	I5	
46-50	I5	
51-55	I5	
56-60	I5	
61-65	I5	
66-70	I5	
71-75	I5	
76-80	I5	Temperature/Pressure Code (See Page 4.0-2)

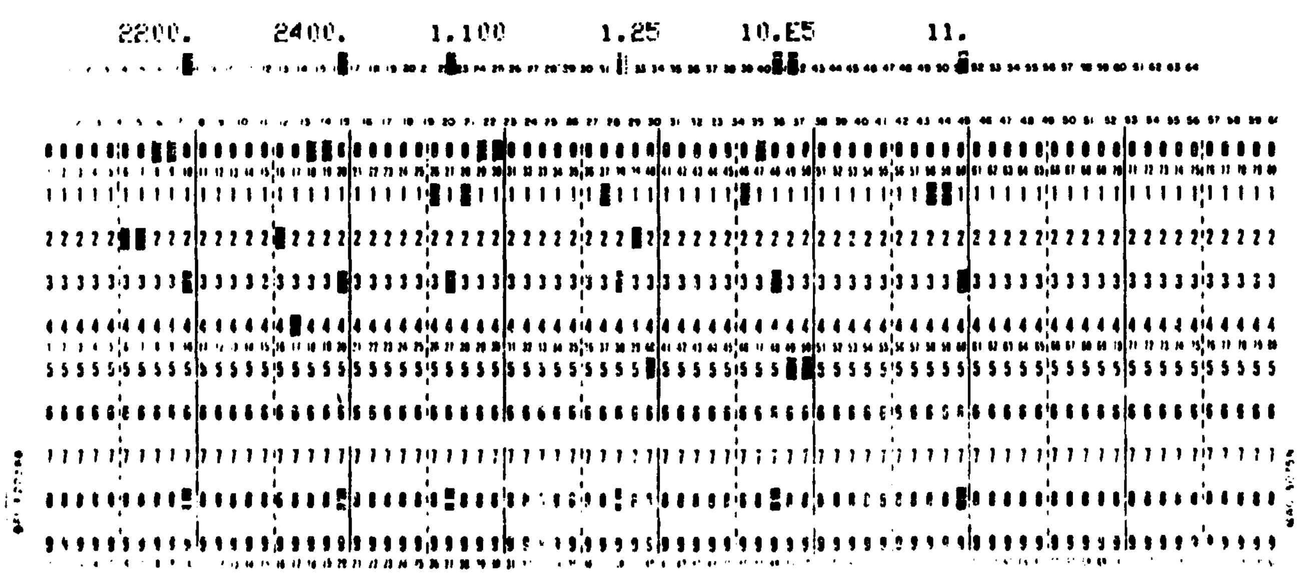
EXAMPLE CARD



CARD NUMBER 2

COLUMN	FORMAT	DATA	DIMENSIONS
1-10	E10.0	Relief Valve Cracking Pressure	PSI
11-20	E10.0	Relief Valve Reseat Pressure	PSI
21-30	E10.0	Slope of Relief Valve Flow Characteristic	PSI/CIS
31-40	E10.0	Slope of Check Valve Flow Characteristic	PSI/CIS
41-50	E10.0	Leakage Impedance	PSI/CIS
51-60	E10.0	Check Valve Cracking Pressure	PSI
61-70	E10.0		
71-80	E10.0		

EXAMPLE CARD



This subroutine does not model the spring/mass dynamics of either the check valve or relief valve. Instead, it establishes an opened or closed condition for each valve, then calculates a component flow based on input data and line pressures supplied by the main program. The pressure at Connection #3, used in an actual priority valve as a reference pressure for the relief valve, is not used at all in subroutine CREL32. This approach yields a very simple model that probably won't give accurate results when hit by transient pressures around its cracking and reseat conditions. Its main use is to simulate gradual shutdown of a low priority function to maintain pressure requirements upstream of the valve.

6.32.1 MATH MODEL

The math model of subroutine CREL32 accounts for five different operating conditions of a priority valve. The conditions and their solution approaches are:

(1) P2 > P1 and P2 > relief valve cracking pressure and P2 > relief valve reseat pressure - the relief valve is fully open, flow is calculated using relief valve flow/ΔP relationship and line pressures.

(2) P2 > P1 and P2 < relief valve reseat pressure but P2 > relief valve cracking pressure with flow already established - the relief valve is open and flow is calculated as in Case (1).

(3) $P2 > P1$ and $P2 <$ relief valve cracking pressure with flow not established - the relief valve is closed, flow is calculated using leakage impedance and line flows.

(4) $P1 > P2$ and $P1-P2 >$ check valve cracking pressure - the check valve is open, reverse flow is calculated using the line pressures and flow/ΔP relationship of the check valve.

(5) $P1 > P2$ and $P1-P2 <$ check valve cracking pressure - the check valve is closed and reverse flow is calculated using line pressures and leakage impedance.

Section 1500

The steady state calculations of subroutine CREL32 consist only of the addition of a flow or leakage term (depending on whether the valve is opened or closed) to the Q leg column and the calculation of the upstream node pressure based on the leg constants and current flow guess.

Section 3000

In the transient section calculations, pressure differential and flow conditions are established to determine which of the cases listed above exists at the particular time step. With this known, a correct flow or leakage impedance can be selected from the input data and used to calculate the flow at Connection #1 as:

$$Q1 = (C1-C2) / (Z1 + Z2 + ZI)$$

Where Q1 = Flow at Connection #1 (CIS)

C1 = Connection #1 Line Characteristic (PSI)

C2 = Connection #2 Line Characteristic (PSI)

Z1 = Connection #1 Line Impedance (PSI/CIS)

Z2 = Connection #2 Line Impedance (PSI/CIS)

ZI = Selected Priority Valve Impedance (PSI/CIS)

Conservation of flow gives the flow at Connections #2 and #3 as:

$$Q2 = -Q1$$

$$Q3 = 0.0$$

Using the basic line equation, the line pressures can be calculated as:

$$P1 = C1 - Q1*Z1$$

$$P2 = C2 - Q2*Z2$$

$$P3 = C3$$

Where

P1 = Pressure at Connection #1 (PSI)

P2 = Pressure at Connection #2 (PSI)

P3 = Pressure at Connection #3 (PSI)

Q2 = Flow at Connection #2 (CIS)

Q3 = Flow at Connection #3 (CIS)

6.32.2 ASSUMPTIONS

Both the relief valve and check valve are assumed to act instantaneously and to be either fully open or fully closed.

If the relief valve is closed for steady state initialization, leakage flow is assumed to allow the flows and pressures to balance.

The reseat pressure of the check valve is assumed to be the same as its cracking pressure.

6.32.3 COMPUTATIONS

1000 Section

Integer codes necessary to describe the component's orientation and operating mode for the steady state initialization are set, then control is returned to the main program.

1500 Section

Tests are made to determine whether relief valve or leakage impedance should be used in the leg calculations. After these tests, the appropriate valve flow factor is selected from the input data and added to the Q leg impedance term. Upstream node pressure is recalculated then control is returned.

2000 Section

DT variables necessary to begin the transient section calculations are initialized.

3000 Section

Connection pressures supplied by the main program are compared to input values of cracking and reseat pressures to determine which mode of operation the priority valve is in. Operating codes are set to record which mode the component is in, and the appropriate component impedance is selected from the input data and used in the calculation of Connection #1 flow. The basic line equations and boundary conditions are then used to give the remaining connection flows and pressures.

6.32.4 APPROXIMATIONS

None.

6.32.5 LIMITATIONS

The CREL32 subroutine does not include the inertia or damping effects of the valve components and does not have a cavitation simulation. Use of CREL32 when these effects are important will give erroneous results.

6.32.6 Variable Names

NAME	DESCRIPTION	DIMENSIONS
D(CLOS)	Relief Valve Reseat Pressure	PSI
D(IPCH)	Check Valve Cracking Pressure	PSI
D(IPREL)	Relief Valve Cracking Pressure	PSI
D(IZCH)	Check Valve Flow Characteristic	PSI/CIS
D(IZLK)	Leakage Impedance	PSI/CIS
D(IZREL)	Relief Valve Flow Characteristic	PSI/CIS
DELP1	Pressure at Inlet (Connection #2)	PSI
DELP2	Component Pressure Drop	PSI
DIMP	Temporary Flow Characteristic	PSI/CIS
DT(P1)	Outlet Pressure (Connection #1)	PSI
DT(P2)	Inlet Pressure (Connection #2)	PSI
DT(P3)	Return Pressure (Connection #3)	PSI
DT(POSO)	Flow Direction Indicator	--
DT(ZV)	Flow Indicator	--
N	Downstream Node Number	--
NPOS1	Cracked Relief Valve Indicator	--
NPOS2	Seated Relief Valve Indicator	--
NPOSCK	Cracked Check Valve Indicator	--
Q1	Component Steady State Flow Value	CIS
ZI	Temporary Relief Valve Flow Characteristic	PSI/CIS

6.32.7 Subroutine Listing

```
      SUBROUTINE CREL32 (D,DT,DD,L)
C *** REVISED DECEMBER 1979 ***
      DOUBLE PRECISION DD
      DIMENSION D(6),DT(5),DD(1),L(4)
      COMMON NTELPL,NTOLPL,IPT,IPOINT,NPTS,INEL,KNEL,NTOPL,NLPLT(61,3),
     1 PQLEG(90,12),LCS(90,10),ILEG(1400),PN(90),QN(90),PEX(90)
      COMMON/SUB/PARM(150,9),PM(1500),QM(1500),P(300),Q(300),C(300)
     1,Z(300),RHO(20),S2ORHO(20),VISC(20),BULK(20),TEMP(20),PVAP(20)
     2,ATPRES,T,DELT,TFINAL,PLTDEL,PI,TITLE(20),LEGN,ICON
     3,KTEMP(99),LSTART(150),NLPT(150),LTYPE(99),NC(99),INX,INZ
     4,INV,ISTEP,NLINE,NEL,IND,IENTR,MNLINE,MNEL,MNLEG,MNNODE,MNPLOT
     5,MNLPTS,MDS
      INTEGER POSO,CLOS,P1,P2,P3,ZV
      DATA IPREL/1/,CLOS/2/,IZREL/3/,IZCH/4/,IZLK/5/,IPCH/6/,POSO/1/
     1 ,P1/2/,P2/3/,P3/4/,ZV/5/
      IF(IENTR) 1000,2000,3000
 1000 CONTINUE
      DT(POSO)=1.0
      IF (INEL.NE.0) GO TO 1500
      IF(L(4).EQ.0) GO TO 1060
      IF(INV) 1030,1050,1040
 1030 IF(INV.EQ.-L(4)) GO TO 1045
      GO TO 1050
 1040 IF(INV.EQ.L(4)) GO TO 1050
 1045 L(4)=1
      GO TO 1060
 1050 L(4)=0
 1060 CONTINUE
      RETURN
C
C     STEADY STATE SECTION
C1500 IF (KNEL-2) 1600,1700,1700
 1500 CONTINUE
 1600 DIMP=D(IZREL)
      IF(L(4).EQ.0) GO TO 1610
      DIMP=D(IZLK)
      DT(POSO)=-1.0
      N=LCS(INEL,3)
      PEX(N)=20.
      QN(N)=D(CLOS)*20.-PQLEG(INEL,1)*PQLEG(INEL,2)
      LCS(INEL,7)=5
 1610 PQLEG(INEL,6)=DIMP+PQLEG(INEL,6)
      QI=PQLEG(INEL,1)
C     DT(P1)=PQLEG(INEL,11)
      PQLEG(INEL,11)=PQLEG(INEL,11)-DIMP*QI*PQLEG(INEL,2)
C     DT(P2)=PQLEG(INEL,11)
      DT(ZV)=DIMP
      LCS(INEL,7)=5
 1700 RETURN
 2000 CONTINUE
```

6.32.7 (Continued)

```
      DT(P2)=P(L(2))
      DT(P1)=P(L(1))
      DT(ZV)=1.001
      RETURN
 3000 CONTINUE
      L1=L(1)
      L2=L(2)
      L3=L(3)
      Q(L3)=0.0
      P(L3)=C(L3)
      DT(P3)=P(L3)
      DELP1=DT(P2)
      DELP2=DT(P1)-DT(P2)
      NPOS1=DELP1/D(IPREL)
      NPOS2=DELP1/D(CLOS)
      NPOSCK=DELP2/D(IPCH)
      IF(DT(POSO))3100,3200,3300
 3100 IF(NPOS1)3200,3130,3120
 3120 ZI=D(IZREL)*SQRT(DT(ZV))
      DT(ZV)=SQRT(DT(ZV))
      IF(DT(ZV).LT.1.001)DT(ZV)=1.001
      DT(POSO)=1.0
      GO TO 3400
 3130 DT(POSO)=-1.0
      IF(NPOSCK)3140,3140,3150
 3140 ZI=D(IZLK)
      GO TO 3400
 3150 ZI=D(IZCH)
      GO TO 3400
 3200 WRITE(6,999)
  999 FORMAT(10X,33HPROGRAM STOP IN SUBROUTINE CREL32)

      STOP 32
 3300 IF (NPOS1)3200,3320,3310
 3310 ZI=D(IZREL)*SQRT(DT(ZV))
      DT(ZV)=SQRT(DT(ZV))
      IF(DT(ZV).LT.1.001)DT(ZV)=1.001
      DT(POSO)=1.0
      GO TO 3400
 3320 IF(NPOS2)3200,3340,3330
 3330 ZI=D(IZREL)*DT(ZV)**2
      DT(POSO)=1.0
      GO TO 3350
 3340 ZI=D(IZREL)*DT(ZV)**2
      DT(POSO)=1.0
      IF(ZI.GT.1100.)DT(POSO)=-1.0
      GO TO 3350
 3350 DT(ZV)=DT(ZV)**2
      IF(DT(ZV).GT.26612.)DT(ZV)=26612.
 3400 Q(L1)=(C(L1)-C(L2))/(Z(L1)+Z(L2)+ZI)
```

6.32.7 (Continued)

```
      Q(L2)=-Q(L1)
      P(L1)=C(L1)-Q(L1)*Z(L1)
      P(L2)=C(L2)-Q(L2)*Z(L2)
      DT(P1)=P(L1)
      DT(P2)=P(L2)
      DT(P3)=P(L3)
C     WRITE(6,4010)NPOS1,NPOS2,NPOSCK,DT(POSO),DELP1,DELP2,ZI
C4010 FORMAT(2X,6HNPOS1=,I4,2X,6HNPOS2=,I4,2X,7HNPOSCK=,I4,2X,4E12.4)
C     WRITE(6,4000) P(L1),Q(L1),P(L2),Q(L2),ZI,C(L1),C(L2),C(L3)
C4000 FORMAT(1X,8E12.4)
      RETURN
      END
```

APPENDIX H

TYPE #41 TWO-WAY ORIFICE RESTRICTORS

HYTRAN USER MANUAL (AFAPL-TR-76-43, VOL. I)

6.41 TYPE #41 TWO-WAY ORIFICE RESTRICTORS

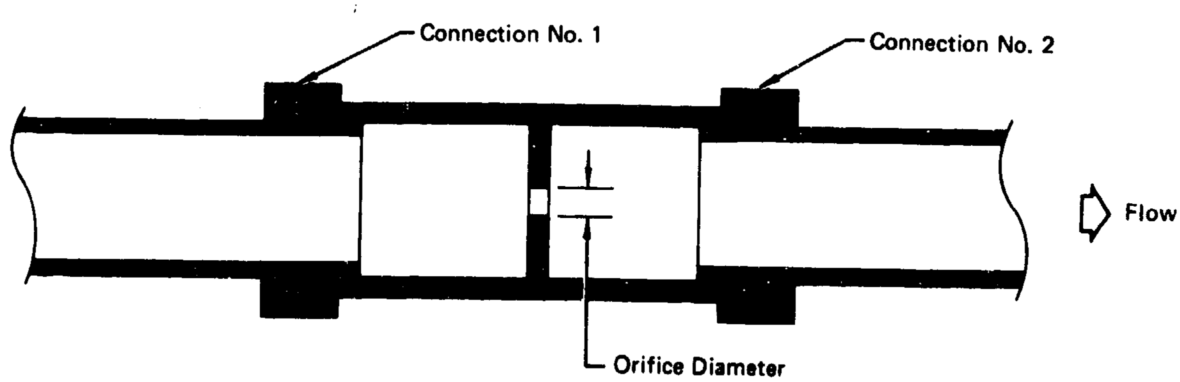


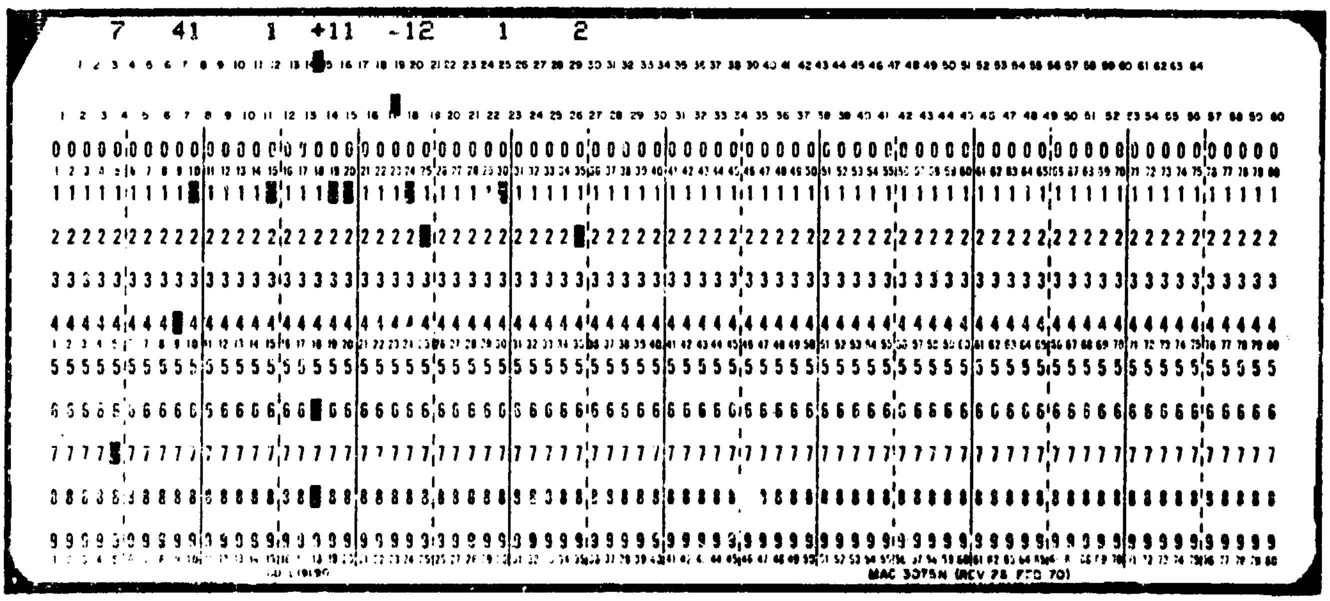
FIGURE 6.41-1
TYPE NO. 41 ORIFICE RESTRICTOR

Type #41 two way restrictors need the line connections and either orifice or rated flow data. If orifice data is selected, the orifice diameter and its discharge coefficient are input. If rated flow data is chosen, the rated flow and pressure drop with the rated fluid and temperature are necessary. Orifice characteristics are assumed to be the same for flow in either direction, so either end may be assigned connection #1.

CARD NUMBER 1

COLUMN	FORMAT	DATA
1-5	I5	Component Number
6-10	I5	Type Number = 41
11-15	I5	Number of Real Data Cards = 1
16-20	I5	Line Number (with sign) attached to Connection 1
21-25	I5	Line Number (with sign) attached to Connection 2
26-30	I5	Type of data input (1=rated flow, 0 or default = orifice)
31-35	I5	Rating fluid (1=MIL-H-5606B, 2=MIL-H-83282, 3=Skydrol)
36-40	I5	
41-45	I5	
46-50	I5	
51-55	I5	
56-60	I5	
61-65	I5	
66-70	I5	
71-75	I5	
76-80	I5	Temperature/Pressure Code (See Page 4.0-2)

EXAMPLE CARD

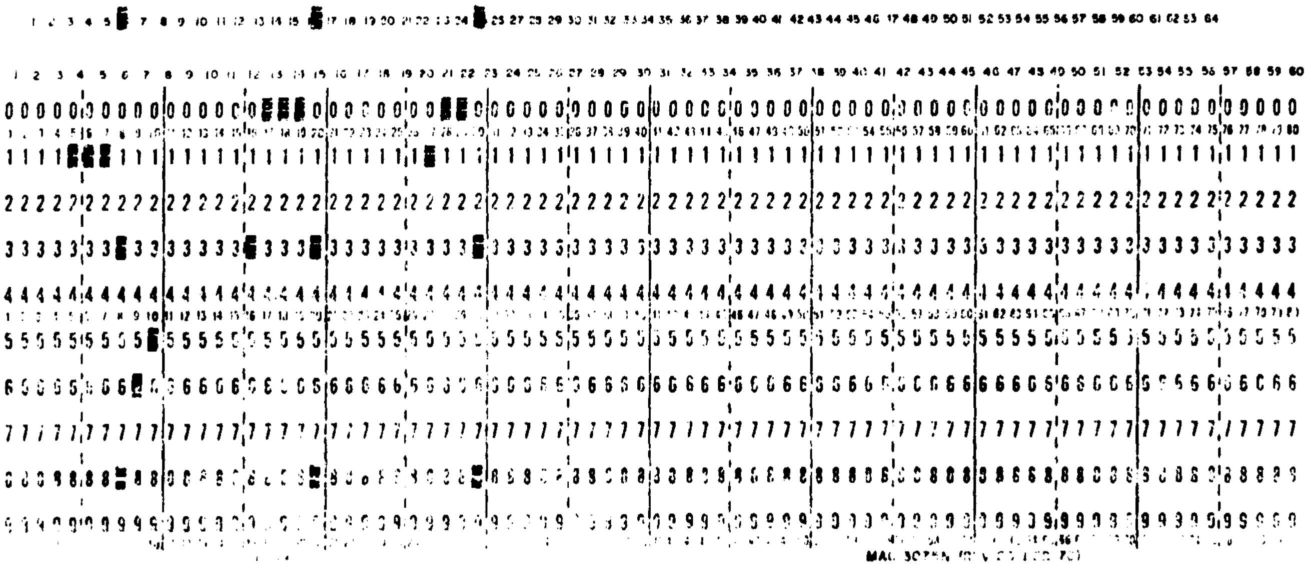


CARD NUMBER 2

COLUMN	FORMAT	DATA	DIMENSIONS
1-10	E10.0	Orifice diameter or rated flow	IN or CIS
11-20	E10.0	Coefficient of discharge or rated DP	- or PSID
21-30	E10.0	Rating fluid temperature	DEGREES F
31-40	E10.0		
41-50	E10.0		
51-60	E10.0		
61-70	E10.0		
71-80	E10.0		

EXAMPLE CARD

111.65 3000. 100.



6.41 SUBROUTINE REST41

REST41 simulates a fixed, two way, orifice type restrictor with two connections. Flow characteristics are assumed to be identical in either direction, so there is no mandatory orientation for the component. Input data may consist of physical parameters (orifice diameter, discharge coefficient) or rated conditions (flow, pressure drop).

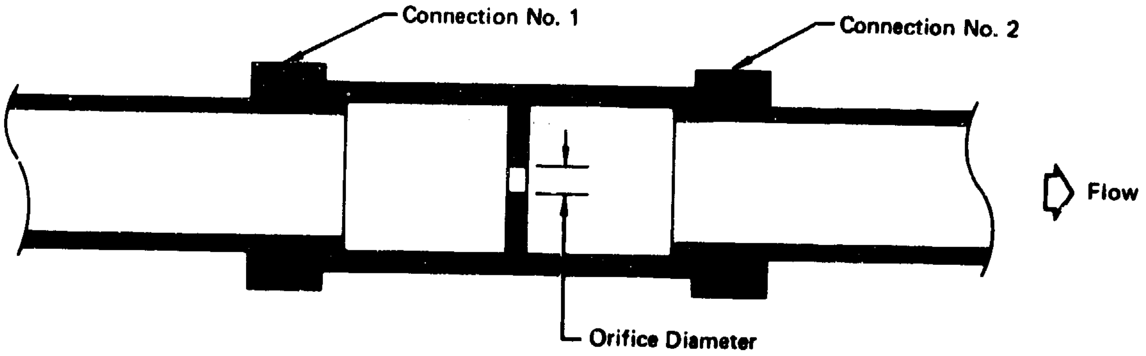


FIGURE 6.41-1
TYPE NO. 41 ORIFICE RESTRICTOR

6.41.1 MATH MODEL FOR NORMAL FLOW

The basic equation for flow through an orifice is:

$$Q = AV*CV*(2*(P1-P2)/RHO)^{1/2} \tag{1}$$

where

AV = area of the valve orifice (in^2)

CV = orifice discharge coefficient

RHO = fluid density ($lb*sec^2/in^4$)

Q = flow (CIS)

P1 = inlet pressure (PSI)

P2 = outlet pressure (PSI)

From the line equations:

$$P1 = C1-Z1*Q1 \tag{2}$$

$$P2 = C2+Z2*Q2 \tag{3}$$

using a positive flow convention from P1 to P2

where C1 = connection #1 line characteristic (PSI)

C2 = connection #2 line characteristic (PSI)

Z1 = connection #1 line impedance (PSI/CIS)

Z2 = connection #2 line impedance (PSI/CIS)

and assuming no flow loss,

$$Q1 = Q2 = Q \tag{4}$$

substituting equations (2) and (3) into equation (1) yields

$$Q^2=AV^2+CV^2*2/RHO \ * \ (C1-C2-Q(Z1+Z2)) \tag{5}$$

By defining $$ZTOT = Z1*Z2 \tag{6}$$

$$CDIFF= C1-C2 \tag{7}$$

$$CONST = AV^2*CV^2*2/RHO \tag{8}$$

equation (5) becomes

$$Q^2+Q*ZTOT*CONST-CONST*CDIFF = 0 \tag{9}$$

which is a quadratic equation with the solution:

$$Q=SIGN \frac{CONST*ZTOT}{2} *(SQRT(1*4*ABS(CDIFF)/(CONST*ZTOT^2))-1, CDIFF) \tag{10}$$

With the flow from equation (10), the line equations (2) and (3) will give the upstream and downstream pressures. Equation (10) is valid for either the rated condition input (where Q, P1 and P2 are given) or the orifice dimension type of input (where AV and CV are given).

6.41.2 CAVITATION MATH MODEL

The cavitation model in REST41 deals with the case where pressure on either one or both sides of the component drops to the vapor pressure of the fluid. When this happens, a bubble or cavity forms in the line and the assumption of zero flow loss is no longer valid. Use of the math model outlined in 6.41.1 would result in the calculation of negative pressures when cavitation occurs. In view of this, some changes to the model are necessary.

CAVITATION AT BOTH CONNECTIONS

When cavitation occurs on both sides of the restrictor, there is no pressure drop across the orifice, and the boundary conditions are:

$$P1 = PVAP$$

$$P2 = PVAP$$

$$QV = 0$$

where

$$P1 = \text{inlet pressure (PSI)}$$

$$P2 = \text{outlet pressure (PSI)}$$

$$QV = \text{flow (CIS)}$$

$$PVAP = \text{vapor pressure of fluid (PSI)}$$

For this case, there may be flow in the upstream and downstream lines. The disparity in flows causes cavitation. Rearrangement of equations (2) and (3) allows calculation of the flows

$$Q1 = (C1-P1)/Z1$$

$$Q2 = (P2-C2)/Z2$$

Formation and collapse of the cavitation bubbles is maintained with DT variables until normal flow is resumed. This is done with the equations

$$DT(CAV1) = DT(CAV1) + QV-Q(L1) \qquad (11)$$

$$DT(CAV2) = DT(CAV2) - QV-Q(L2) \qquad (12)$$

where

DT(CAV1) = flow discrepancy at connection 1 (CIS)

DT(CAV2) = flow discrepancy at connection 2 (CIS)

QV = flow through orifice (CIS)

Q(L1) = flow in line connected to connection 1 (CIS)

Q(L2) = flow in line connected to connection 2 (CIS)

CAVITATION AT DOWNSTREAM CONNECTION

In this case, a cavitation bubble forms downstream of the restrictor. There is flow through the orifice, but flow in the upstream and downstream lines is not the same because of the bubble formation.

Beginning with equation (1) for flow through an orifice, the line equation for the upstream line (equation (2)), and the boundary condition:

$$P2 = PVAP$$

Equation (1) is rewritten as:

$$Q^2 = AV^2*CV^2*2/RHO*(C1-Z1*Q1-PVAP) \tag{13}$$

by defining $CONST = AV^2*CV^2*2/RHO$

$CDIFF = C1-PVAP$

equation (13) simplifies to

$$Q^2 = CONST*(CDIFF-Z1*Q)$$

which is the same as equation (9) with the same solution (Equation 10)

Use of the line equations and boundary conditions gives the line pressures and flows as

$$Q1 = Q$$

$$P1 = C1-Q1/Z1$$

$$P2 = PVAP$$

$$Q2 = (C2-P2)/Z2$$

Flow discrepancy is monitored by equation (12).

CAVITATION AT UPSTREAM CONNECTION

In this case the cavitation bubble forms upstream of the orifice. The basic flow equation (1) holds, but the line equation for the downstream line (Equation (3)) is used and the boundary condition (P1 = PVAP) is applied to the upstream side of the component.

When inserted into the flow equation (1), this yields

$$Q^2 = AV^2*CV^2*2/RHO*(PVAP-(C2+Q*Z2)) \tag{14}$$

with $CONST = AV^2*CV^2*2/RHO$

$CDIFF = PVAP-C2$

Equation (14) simplifies to

$$Q^2 = CONST*(CDIFF-Q2*Z2)$$

which again is the same as equation (9) and has the same solution. Applying the line equation and boundary condition gives:

P1 = PVAP

Q1 = (C1-P1)/Z1

P2 = C2+Q*Z2

Q2 = Q

The bubble is recorded and updated by equation (11) in this case.

6.41.3 ASSUMPTIONS

1. The restrictor has no ancillary parts and the oil volume is sufficiently small so integration is not required.

2. The discharge coefficient is the same for flow in either direction and is constant over the entire flow regime.

6.41.4 COMPUTATION METHODS

SECTION 1000

A check is made for the type of input being used. If the orifice type input is used, the area of the orifice is calculated from the input diameter, then used with the discharge coefficient to establish the orifice characteristic.

$$RV*AREA*COEFF*\sqrt{2/RHO}$$

If the rated flow input is used, the rated fluid and temperature are determined. The density of the rated fluid at the rated temperature is calculated. Default conditions will establish MIL-H-5606 fluid @ 100°F as the rated fluid. Once the density of the rated fluid is calculated, a density correction factor is computed using the fluid density for the component.

DENSC = SQRT (RATED DENSITY/SIMULATION DENSITY)

The orifice characteristic is determined using the rated flow, pressure drop, and density correction factor.

RV=DENSC*RATED FLOW/SQRT (RATED PRESSURE DROP)

SECTION 1500

Using the orifice characteristic from the 1000 section, the leg constant for Q^2 is updated and pressure at the outlet connection is calculated and stored.

SECTION 2000

No calculations are performed. Control is returned to the main program.

SECTION 3000

A check is made to determine if the component is in a cavitation situation. If it is not, flow is calculated using equation (10). The flow is used to calculate pressures P1 and P2. A check is made to determine if P1 or P2 is less than the fluid vapor pressure. If neither is, control is returned to the main program.

If the component is cavitated, flow through the orifice is calculated using equation (10) except that ZT and CDIFF are changed, depending on the location of cavitation. With the flow determined, use of the appropriate line equations and boundary conditions gives the values of P1 and P2, and allows calculation of flow discontinuity. DT variables keep track of the flow difference until the cavitation stops and normal flow resumes. The actual boundary conditions and attendant equations for each type of cavitation are outlined in Section 6.41.2.

6.41.5 LIMITATIONS

Subroutine REST41 is limited to a fixed orifice two-way restrictor with the same discharge coefficient for flow in either direction.

6.41.6 Variable Names

NAME	DESCRIPTION	DIMENSIONS
CDIFF	Characteristic Pressure Difference	PSI
D(DATA1)	Orifice Diameter or Rate Flow	IN or CIS
D(DATA2)	Discharge Coefficient or Rated Pressure Drop	- or PSI
DENS	Rated Fluid Density	$LB*SEC^2/IN^4$
DENSC	Density Correction Factor	--
DT(CAV1)	Connection #1 Cavitation Monitor	CIS
DT(CAV2)	Connection #2 Cavitation Monitor	CIS
DT(RV)	Orifice Characteristic	$CIS/\sqrt{PSI}$
P1	Connection #1 Cavitation Pressure	PSI
P2	Connection #2 Cavitation Pressure	PSI
PV	Fluid Vapor Pressure	PSI
Q1	Normal Orifice Flow	CIS
QV	Cavitated Orifice Flow	CIS
RFLUID	Rated Fluid Code	--
RTEMP	Rated Temperature	DEGREES F
ZT	Total Line Impedance	PSI/CIS

6.41.7 Subroutine Listing

```
      SUBROUTINE REST41 (D,DT,DD,L)
C *** REVISED DECEMBER 1979 ***
C *** THIS SUBROUTINE INCORPORATES A CAVITATION MODEL ON THE
C *** UPSTREAM AND DOWNSTREAM SIDE OF THE RESTRICTOR. IT ALSO INCLUDES
C *** THE OPTION OF SPECIFYING VALVE CHARACTERISTICS BY ORIFICE
C *** DIAMETER AND COEFFICIENT OF DISCHARGE OR BY RATED FLOW AND
C *** PRESSURE DROP.
      DOUBLE PRECISION DD
      DIMENSION D(4),DT(2),DD(1),L(4)
      COMMON NTELPL,NTOLPL,IPT,IPOINT,NPTS,INEL,KNEL,NTOPL,NLPLT(61,3),
     1 PQLEG(90,12),LCS(90,10),ILEG(1400),PN(90),QN(90)
      COMMON/SUB/PARM(150,9),PM(1500),QM(1500),P(300),Q(300),C(300)
     1,Z(300),RHO(20),S2ORHO(20),VISC(20),BULK(20),TEMP(20),PVAP(20)
     2,ATPRES,T,DELT,TFINAL,PLADEL,PI,TITLE(20),LEGN,ICON
     3,KTEMP(99),LSTART(150),NLPT(150),LTYPE(99),NC(99),INX,INZ
     4,INV,ISTEP,NLINE,NEL,IND,IENTR,MNLINE,MNEL,MNLEG,MNNODE,MNPLOT
     5,MNLPTS,MDS
      INTEGER DATA1,DATA2,RV,CAV1,CAV2,RFLUID,RTEMP
C ****** D VARIABLES ********
      DATA DATA1/1/,DATA2/2/,RTEMP/3/
C ***** DT VARIABLES *******
      DATA RV/1/,CAV1/2/,CAV2/3/
      IF(IENTR) 1000,2000,3000
 1000 CONTINUE
      IF (INEL.NE.0) GO TO 1500
      IF(L(3).NE.0) GO TO 1001
C *** CALCULATIONS FOR ORIFICE / COEFFICIENT OF DISCHARGE INPUT
      D(DATA1)=D(DATA1)**2*PI/4.0
      DT(RV)=D(DATA1)*D(DATA2)*S2ORHO(KTEMP(IND))
      GO TO 1002
C *** CALCULATIONS FOR RATED CONDITION INPUT
 1001 IF(L(4).EQ.0) GO TO 1003
C *** DENSITY CALCULATION FOR RATED CONDITIONS
      RFLUID=L(4)
      IF(RFLUID-2)1004,1005,1006
C *** DENSITY CALCULATION OF MIL-H-5606B AT RATED TEMPERATURE
 1004 DENS=56.0147-1.82353E-2*RTEMP
      GO TO 1007
C *** DENSITY CALCULATION OF MIL-H-83282 AT RATED TEMPERATURE
 1005 DENS=55.0706-2.35294E-2*RTEMP
      GO TO 1007
C *** DENSITY CALCULATION OF SKYDROL 500B AT RATED TEMPERATURE
 1006 DENS=67.0221-2.73529E-2*RTEMP
      GO TO 1007
C *** DEFAULT DENSITY
 1003 DENS=54.19117
C *** CHANGE TO CORRECT DENSITY UNITS
 1007 DENS=DENS/(12.**4.*32.174)
C *** DENSITY CORRECTION TO COMPONENT CONDITIONS
      DENSC=SQRT(DENS/RHO(KTEMP(IND)))
```

6.41.7 (Continued)

```
      DT(RV)=D(DATA1)/SQRT(D(DATA2))*DENSC
 1002 DT(CAV1)=0.0
      DT(CAV2)=0.0
      RETURN
C
C     STEADY STATE SECTION
 1500 CONTINUE
      PQLEG(INEL,8)=PQLEG(INEL,8)+1./DT(RV)**2
      QI=PQLEG(INEL,1)
      PQLEG(INEL,11)=PQLEG(INEL,11)-PQLEG(INEL,2)*QI**2/DT(RV)**2
      RETURN
 2000 CONTINUE
      RETURN
 3000 CONTINUE
      L1=L(1)
      L2=L(2)
      PV=PVAP(KTEMP(IND))
      IF(DT(CAV1)+DT(CAV2).GT.0.0) GO TO 3300
      CDIFF=C(L1)-C(L2)
      ZT=Z(L1)+Z(L2)
      QI=SIGN((DT(RV)**2*ZT/2.)*(SQRT(1.+4.*ABS(CDIFF)/(DT(RV)**2
     +*ZT**2))-1.),CDIFF)
      P(L1)=C(L1)-QI*Z(L1)
      Q(L1)=QI
      P(L2)=C(L2)+QI*Z(L2)
      Q(L2)=-QI
      IF(P(L1).GE.PV.AND.P(L2).GE.PV)RETURN
      IF(P(L1).LT.PV.AND.P(L2).LT.PV) GO TO 3230
      IF(P(L1).LT.PV) GO TO 3220
 3210 P2=PV
      ZT=Z(L1)
      P1=C(L1)
      IF(C(L1).GT.PV) GO TO 3500
      P1=PV
      GO TO 3600
 3220 P1=PV
      ZT=Z(L2)
      P2=C(L2)
      IF(C(L2).GT.PV) GO TO 3500
      P2=PV
      GO TO 3600
 3230 IF(P(L1)-P(L2))3210,3400,3220
 3300 IF(DT(CAV1).GT.0.0.AND.DT(CAV2).GT.0.0)GO TO 3400
      IF(DT(CAV1).GT.0.0) GO TO 3220
      GO TO 3210
 3400 P1=PV
      P2=PV
      GO TO 3600
 3500 QV=SIGN((DT(RV)**2*ZT/2.)*(SQRT(1.+4.*ABS(P1-P2)/(DT(RV)**2
     +*ZT**2))-1.),(P1-P2))
```

6.41.7 (Continued)

```
      IF(P1.GT.PV) GO TO 3700
      GO TO 3800
3600  QV=0.0
      GO TO 3750
3700  P(L1)=C(L1)-QV*Z(L1)
      Q(L1)=QV
3750  Q(L2)=(C(L2)-P2)/Z(L2)
      P(L2)=P2
      DT(CAV2)=DT(CAV2)-QV-Q(L2)
      IF(QV.EQ.0.0) GO TO 3850
      RETURN
3800  P(L2)=C(L2)+QV*Z(L2)
      Q(L2)=QV
3850  Q(L1)=(C(L1)-P1)/Z(L1)
      P(L1)=P1
      DT(CAV1)=DT(CAV1)+QV-Q(L1)
      RETURN
      END
```

APPENDIX I

TYPE #63 RLS BOOTSTRAP RESERVOIR

HYTRAN USER MANUAL (AFAPL-TR-76-43, VOL. I)

6.63 TYPE #63 RLS BOOTSTRAP RESERVOIR

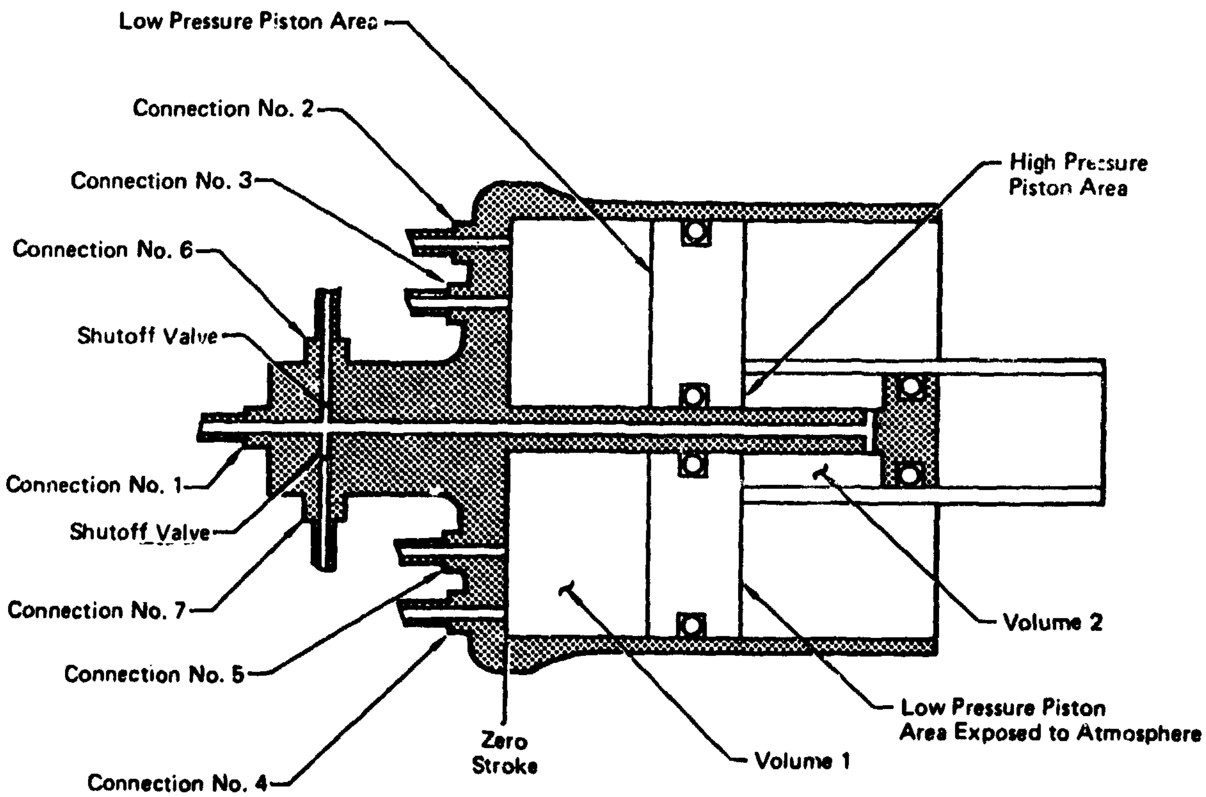


FIGURE 6.63-1
TYPE NO. 63 RLS BOOTSTRAP RESERVOIR

The Type #63 reservoir is a two circuit, level sensing, bootstrap reservoir. Such reservoirs are used on the F-15 and F-18 aircraft to provide protection from total system loss when a leak developes in one of the two independent subcircuits.

This model does not simulate the shut-off characteristics or level sensing ability of the reservoir but does include the pressure drop due to the shut-off valves when they are fully opened.

The RLS bootstrap reservoir can have as many as seven connections. It requires one high pressure inlet, two high pressure outlets and from one to four low pressure connections. Any low pressure connections that are not used must be left blank and the number of unused connections must be input on data card #1.

For steady state balancing, the component requires two nodes. One for the low pressure connections and one for the high pressure connections. When assembling the legs, the leg attached to Connection #1 should be numerically lower than any of the other legs attached to the reservoir.

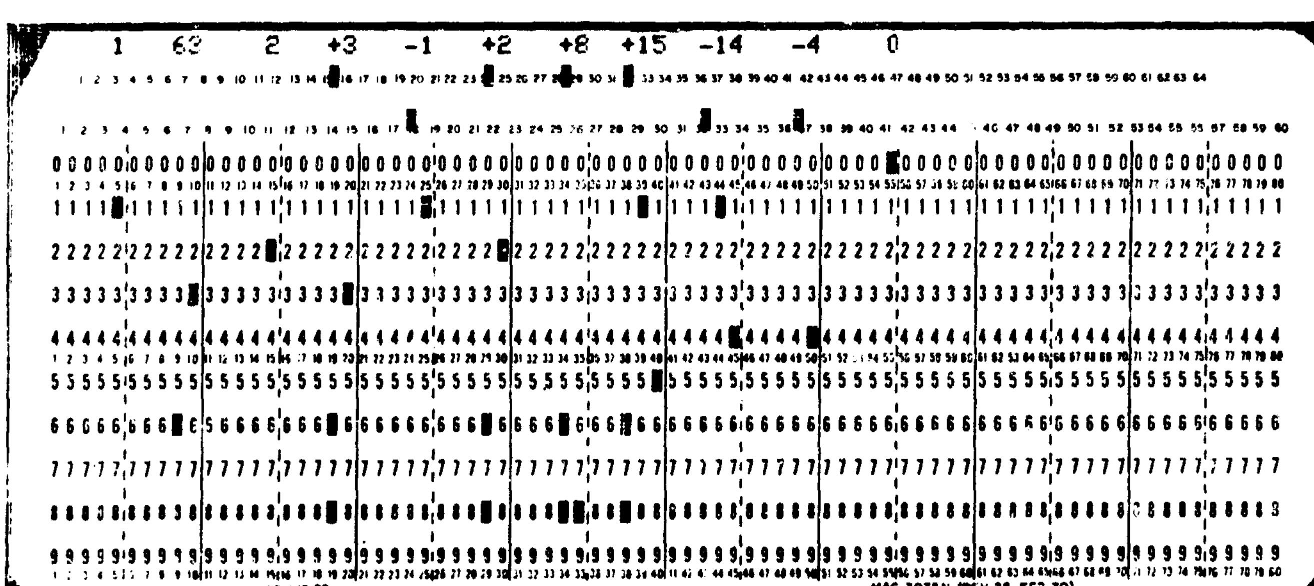
If more than one of these reservoirs is used in a simulation, all of the lines and legs connected to the first reservoir should be numerically lower than the lines and legs connected to the second reservoir.

The model does not simulate cavitation in either the high or low pressure sections.

CARD NUMBER 1

COLUMN	FORMAT	DATA
1-5	I5	Component Number
6-10	I5	Type Number = 63
11-15	I5	Number of Real Data Cards = 2
16-20	I5	Line Number (with sign) attached to Connection 1 (Bootstrap)
21-25	I5	Line Number (with sign) attached to Connection 2
26-30	I5	Line Number (with sign) attached to Connection 3
31-35	I5	Line Number (with sign) attached to Connection 4
36-40	I5	Line Number (with sign) attached to Connection 5
41-45	I5	Line Number (with sign) attached to Connection 6 (Circuit A)
46-50	I5	Line Number (with sign) attached to Connection 7 (Circuit B)
51-55	I5	Number of unused low pressure connections
56-60	I5	
61-65	I5	
66-70	I5	
71-75	I5	
76-80	I5	Temperature/Pressure Code (See Page 4.0-2)

EXAMPLE CARD

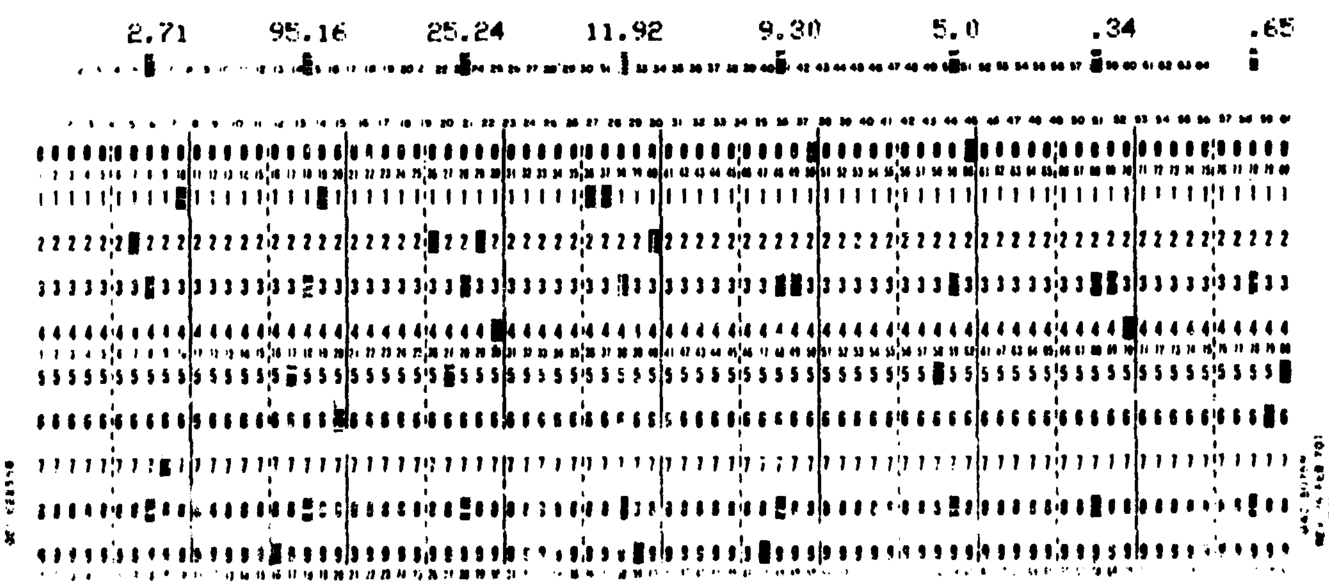


CARD NUMBER 2

COLUMN	FORMAT	DATA	DIMENSIONS
1-10	E10.0	High Pressure (Boostrap) Piston Area	IN^2
11-20	E10.0	Low Pressure (Reservoir) Piston Area	IN^2
21-30	E10.0	High Pressure Volume at Zero Stroke*	IN^3
31-40	E10.0	Low Pressure Volume at Zero Stroke*	IN^3
41-50	E10.0	Maximum Piston Stroke	IN
51-60	E10.0	Initial Piston Position	IN
61-70	E10.0	Shut-Off Valve Orifice Diameter (Circuit A)	IN
71-80	E10.0	Shut-Off Valve Discharge Coefficient(Circuit A)	-

*Note: Zero Stroke is defined as the Piston Position with the Reservoir Empty.

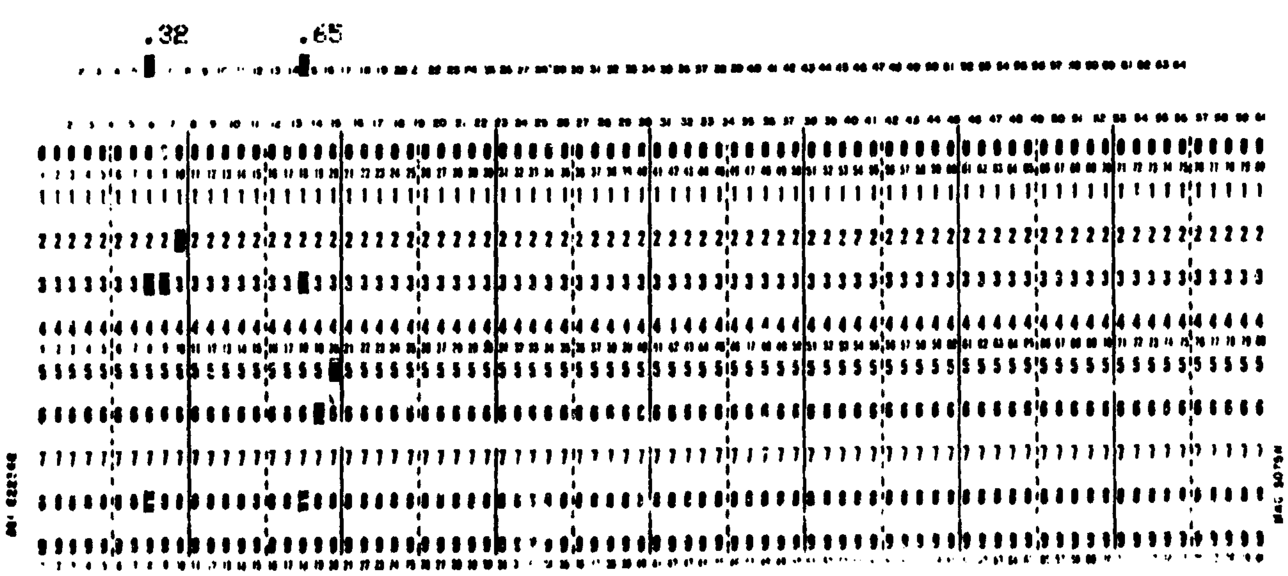
EXAMPLE CARD



CARD NUMBER 3

COLUMN	FORMAT	DATA	DIMENSIONS
1-10	E10.0	Shut-Off Valve Orifice Diameter (Circuit B)	IN
11-20	E10.0	Shut-Off Valve Discharge Coefficient (Circuit B)	-
21-30	E10.0		
31-40	E10.0		
41-50	E10.0		
51-60	E10.0		
61-70	E10.0		
71-80	E10.0		

EXAMPLE CARD



6.63 SUBROUTINE RSVR63

Subroutine RSVR63 is a model of a dual circuit, Level Sensing (RLS) bootstrap reservoir. Such devices are in use on the F-15 and F-18 aircraft and provide protection from total system loss due to a leak in one of the two independent subcircuits.

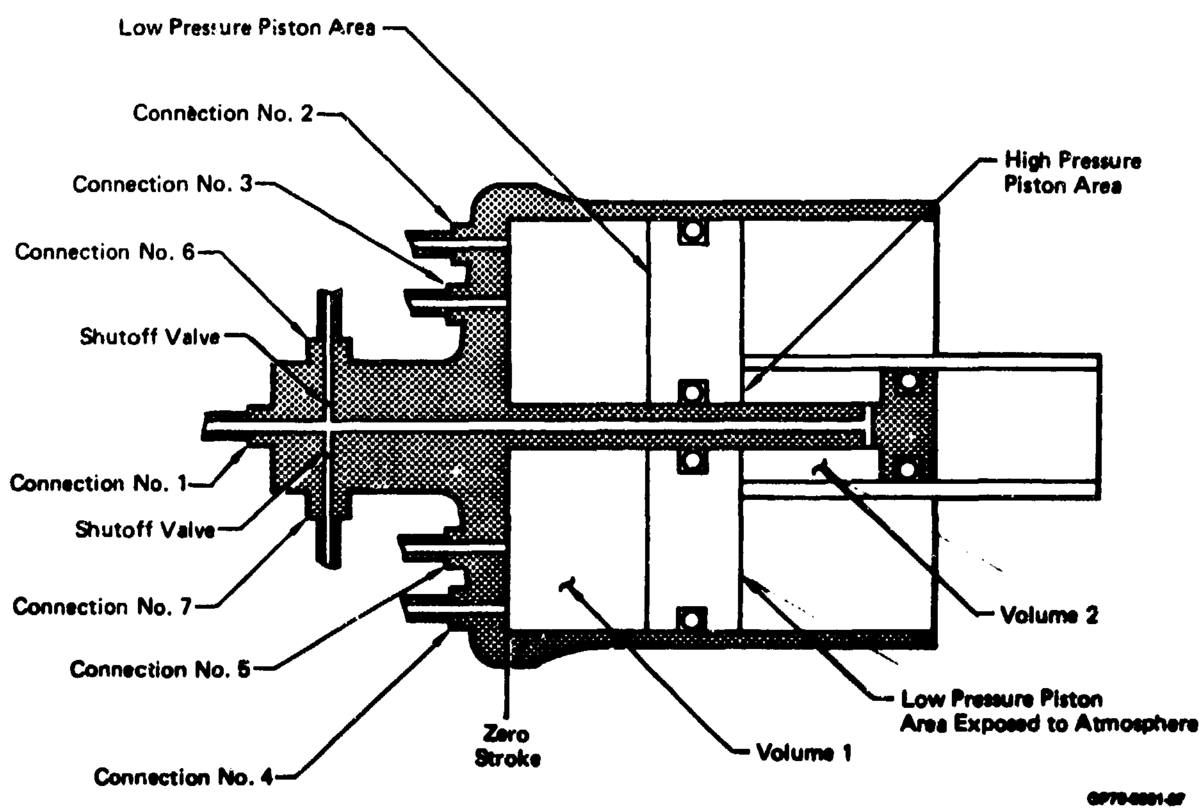


FIGURE 6.63-1
TYPE NO. 63 RLS BOOTSTRAP RESERVOIR

RSVR63 allows a maximum of seven line connections. One high pressure inlet, two high pressure outlets, and up to four low pressure connections. As shown in Figure 6.63-1, the high pressure inlet branch splits to provide bootstrap pressure to the reservoir and flow to the independent supply circuits. This model does not simulate the shutoff characteristics of the RLS valves, but does simulate the pressure drop due to them.

RSVR63 requires two nodes for its steady state calculations. One node is located in the low pressure (reservoir) section and the other node is located in the high pressure (bootstrap) section. When assembling the steady state legs that connect to RSVR63, the leg attached to Connection #1 should be numerically lower than any of the other legs connected to the reservoir.

If more than one of these reservoirs is used in a simulation, the lines of the system should be numbered so that all of the lines connected to the first reservoir are numerically lower than any of the lines connected to the second reservoir.

RSVR63 does not model piston dynamics and will not give good results when cavitated.

6.63.1 MATH MODEL

Since the component is a combination of a reservoir and two parallel restrictor paths, its math model consists of a bootstrap reservoir model and a restrictor model. The restrictor portion of the model is the same as that explained in Section 6.41.1, and is the more independent of the two portions. The reservoir portion of the model is dependent on the restrictor calculations because the bootstrap pressure is determined, in large part, by the flow through the RLS circuits.

1500 Section

In the 1500 section of the RLS reservoir, the bootstrap node pressure is determined by the flow through the RLS circuits. The bootstrap node is the upstream node of the high pressure legs leaving the reservoir. As such, addition of the RLS shutoff valve characteristics to the Q^2 term of the steady state solution allows calculation of the upstream node pressure. The assumption is made that the pressure in the bootstrap chamber of the reservoir is the same as the node pressure.

The problem then becomes one of establishing a pressure at the low pressure node that will allow both a steady state flow balance and a force balance on the reservoir piston. To do this, a sign convention is established such that flow into the low pressure reservoir chamber is positive. A pseudo leg (see Figure 6.63-2) that terminates at the low pressure node (N) is established.

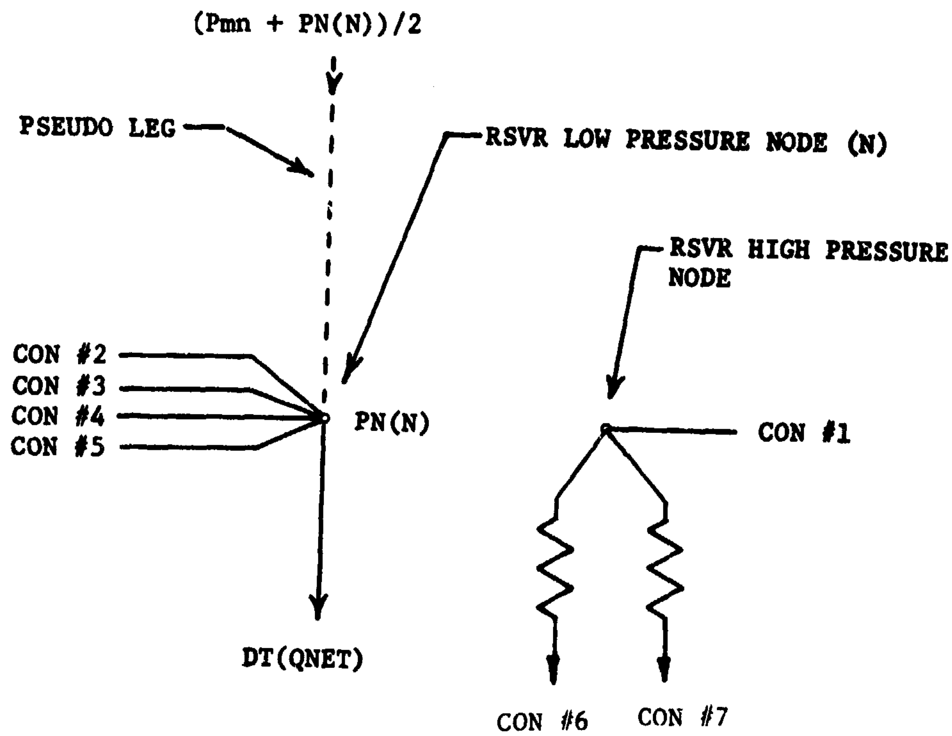


FIGURE 6.63-2

The pressure at the external end of this leg is set as the average of the pressure at node (N) and the pressure necessary to balance the force exerted by the bootstrap pressure and the atmosphere. The balancing pressure is calculated as:

$$P_{mn} = Pm^* \frac{\text{High Pressure Area}}{\text{Low Pressure Area}} + \frac{(\text{Low Pressure Area}) - (\text{High Pressure Area})}{\text{Low Pressure Area}} * \text{Patm}$$

Where $\quad Pmn$ = Pressure Necessary for Balance (PSI)

Pm = Bootstrap Node Pressure (PSI)

$Patm$ = Atmospheric Pressure (PSI)

Any difference in the pressure at node (N) and the value of Pmn will produce flow in the pseudo leg and hence an unbalanced system. As CALC balances the flows at all system nodes, the pseudo leg flow is forced to zero which in turn forces Pmn and the pressure at node (N) to be equal.

3000 Section

The transient section math model of the RLS reservoir is a combined force and flow balance between the high and low pressure sections.

Beginning with the basic equation for any given line, "L",

$$P(L) = C(L) - Z(L) * Q(L)$$

Where $\quad P(L)$ = Pressure of Line "L" (PSI)

$C(L)$ = Characteristic of Line "L" (PSI)

$Z(L)$ = Impedance of Line "L" (PSI/CIS)

$Q(L)$ = Flow in Line "L" (CIS)

and the assumption that the pressure of any line connected to the reservoir (Low Pressure) part of the component is the same as the reservoir chamber pressure, the net flow (into or out of) the reservoir chamber can be expressed as:

$$QN = CN - \left[(C1 - Q1*Z1) * \frac{\text{High Press. Area}}{\text{Low Press. Area}} + \text{Express} \right] * \frac{1}{ZN} \qquad (1)$$

Where $\quad QN$ = Net Flow Into Reservoir Chamber (CIS)

$$CN = \Sigma \left[\frac{\text{Low Pressure Line Characteristic}}{\text{Low Pressure Line Impedance}} \right] \text{(CIS)}$$

$C1$ = Bootstrap Line Flow (CIS)

$Z1$ = Bootstrap Line Impedance (PSI/CIS)

EXPRESS = Pressure due to atmosphere acting on area unbalance (PSI)

ZN = Σ of Low Pressure Line Impedances (PSI/CIS)

The net flow can be related to the flow into or out of the bootstrap chamber through the equation:

$$Q_{BS} = QN* \frac{\text{High Pressure Area}}{\text{Low Pressure Area}} \qquad (2)$$

Use of the restrictor equations, as covered by Section (6.41) will give the flow through both RLS valves. The sum of these flows and Q_{BS} is the flow in line Number 1. If the calculation of Q1 and the value of Q1 supplied by the main program are within ± .001 CIS, pressures and flows may be calculated using the appropriate line equations and area differentials.

If the tolerance test between the main program's value of Q1 and the subroutine's Q1 is not met, Q1 can be redefined as:

$$Q1 = Q1_p + Qtot \ /2. \qquad (3)$$

Where Q1 = Revised Value of Line #1 Flow (CIS)

$Q1_p$ = Value of Q1 supplied by Program (CIS)

Qtot = Value of Q1 calculated by Subroutine (CIS)

= Q6 + Q7 + OBS

An iteration using Equations (1), (2), and (3) can be set up to calculate a value of Q1 that will meet the ± .001 CIS test. Assigning this value as the flow into Connection #1, the line pressure can be calculated using the basic line equation, then the pressure in the reservoir chamber can be calculated using the area ratios. Assigning the reservoir chamber pressure as the line pressure for all lines connected to the reservoir section, the flows in these lines can then be determined using the basic line equation.

6.63.2 ASSUMPTIONS

It is assumed that the piston movements in a reservoir like this are small enough to justify neglecting piston spring-mass effects and seal friction.

6.63.3 COMPUTATION METHODS

Section 1000

Counters and variables that will be necessary in the later sections of the subroutine are initialized and stored. Orifice characteristics of the RLS valves and volume constants that will be needed to check for overflowed and depleted conditions are calculated, then control is returned to the main program.

Section 1500

The steady state calculations begin with a check of which connection number of the reservoir is under consideration. If connections 6 or 7 (the RLS outlets) are in the leg being calculated, the appropriate orifice constant is added to the Q^2 leg term, the upstream node pressure is recalculated then control is returned.

If the high pressure (Bootstrap) connection is in the leg under calculation, tests are performed to determine which node number is located at the bootstrap chamber, then control is returned.

When the low pressure (reservoir) connections are under calculation, a check on which node number is located at the low pressure reservoir chamber is made, then the flow in the leg under calculation is added to the variable DT(QNET). If the connection being calculated is not the last reservoir connection to be called during the iteration, control is returned to the main program.

If the connection is the last reservoir connection to be called, the pseudo node pressure Pmn is calculated using the bootstrap node pressure, differential area and atmospheric pressure. The flow in the pseudo leg is calculated using Pmn, the reservoir node pressure, DT(QNET), and an arbitrarily assigned dummy conductance that is later removed from the calculations. The DT variable P2 is set equal to the reservoir node pressure, and the connection counter is zeroed in preparation for the next iteration before control is relinquished.

Section 2000

This section initializes the variables and establishes the constants that are necessary to begin the transient calculations in Section 3000.

Section 3000

Counters and variables that will be used during the time step are first reset, then the orifice factors necessary for the RLS circuit calculations are established.

The variable CN2 is calculated in order to establish the bootstrap chamber flow which, when added to the RLS circuit flows calculated by the restrictor equation, gives the first estimate of flows into the high pressure inlet (Connection #1). If this flow estimate is within $\pm$ .001 CIS of the Connection #1 line flow supplied by the main program, the flows are accepted as correct and used in the line equations to calculate the pressures at Connections 1, 6 and 7. If the tolerance test is not met, the Connection #1 flow value is averaged with the flow just calculated by the subroutine and a new estimate of Connection #1 flow is made using the revised value. This cycle is continued for 25 iterations or until the flow tolerance test is met.

After the flows and pressures in the high pressure section of the reservoir are established, the area differential, external pressure and pressure at Connection #1 are used to calculate the pressure on the large side of the piston.

Using the new value of net flow, the volume of the reservoir chamber is updated, then checked against its limits to establish a depleted or overflowed condition. If either of these conditions exist, the reservoir pressure (Low Pressure) is recalculated using line characteristics and impedances, then the high pressure section calculations are performed again with the appropriate restrictions.

The final steps of the subroutine are to calculate line flows in the reservoir section based on the pressure acting on the large area, update the net flow and volume DT variables, and reset a counter that will be necessary for the next time step's calculations. Control is then returned to the main program.

6.63.4 APPROXIMATIONS

The elasticity of the reservoir walls, compressibility effects of the fluid in the chambers and volume changes resulting from any piston movements are ignored. There is no pressure drop between the high pressure inlet (Connection #1) and the bootstrap chamber or RLS shut-off valves.

6.63.5 LIMITATIONS

RSVR63 is limited to four low pressure connections, one high pressure inlet and two high pressure outlets.

6.63.6 Variable Names

NAME	DESCRIPTION	DIMENSION
B3	Connection #6 Line Impedance	PSI/CIS
B4	Connection #7 Line Impedance	PSI/CIS
CL1	Connection #1 Pressure Guess	PSI
CN2	Low Pressure Volume Flow Guess	CIS
D1	RLS Circuit Valve Characteristic	PSI/CIS^2
D3	RLS Circuit A Characteristic Pressure	PSI
D4	RLS Circuit B Characteristic Pressure	PSI
D(A1)	RLS Circuit A Orifice Area	IN^2
D(A2)	RLS Circuit B Orifice Area	IN^2
D(AREA1)	High Pressure (Bootstrap) Piston Area	IN^2
D(AREA2)	Low Pressure (Reservoir) Piston Area	IN^2
D(CDA)	Discharge Coefficient of Circuit A Orifice	--
(CDB)	Discharge Coefficient of Circuit B Orifice	--
D(INPOS)	Piston Position at Start of Simulation	IN
D(STROKE)	Total Piston Stroke	IN
D(VOL1)	Bootstrap Chamber Volume at Zero Stroke	IN^3
D(VOL2)	Reservoir Chamber Volume at Zero Stroke (i.e. When fully depleted)	IN^2
DT(DUM)	Reservoir Fluid Volume	IN^3
DT(EXPRESS)	Reservoir Fluid Pressure due to Atmosphere Acting on Area Urbalance	PSI
DT(IQV)	Time Rate of Change of Reservoir Fluid Volume	CIS
DT(K1)	Circuit A Orifice Characteristic	CIS^2/PSI
DT(K2)	Circuit B Orifice Characteristic	CIS^2/PSI

NAME	DESCRIPTION	DIMENSION
DT(MAXIQV)	Volume Rate of Change Limit for Overflow Check	CIS
DT(MINIQV)	Volume Rate of Change Limit for Depletion Check	CIS
DT(NAREAR)	Piston Area Ratio (High PR/Low PR)	--
DT(NZ1)	Pseudo Impedance of Reservoir Volume	PSI/CIS
DT(NZG)	Pseudo Conductance of Reservoir Volume	CIS/PSI
DT(P2)	Reservoir Chamber Pressure	PSI
DT(QNET)	Net Reservoir Chamber Flow	CIS
DT(ZN)	Combined Conductance of Reservoir Section Lines	CIS/PSI
ITER	Iteration Counter	--
ITEST	Iteration Counter	--
PMN	Steady State Reservoir Chamber Pressure	PSI
Q1	Flow Into Connection #1	CIS
Q2	Flow into Bootstrap Chamber	CIS
Q6	RLS Circuit A Flow (Connection #6)	CIS
Q7	RLS Circuit B Flow (Connection #7)	CIS
QA	Steady State Connection #1 Flow	CIS
QR	Flow Vector	CIS
QS	Steady State Connection #1 Flow Sign	--
QTOT	Total of Bootstrap Chamber and RLS Circuit Flows	CIS
R1	RLS Circuit #1 Valve Characteristic	PSI/CIS^2
R2	RLS Circuit #2 Valve Characteristic	PSI/CIS^2
RO	Component Fluid Density	$LB*SEC^2/IN^4$

6.63.7 Subroutine Listing

```
      SUBROUTINE RSVR63 (D,DT,DD,L)
C *** REVISED DECEMBER 1979 ***
      DOUBLE PRECISION DD
      COMMON NTELPL,NTOLPL,IPT,IPOINT,NPTS,INEL,KNEL,NTOPL,NLPLT(61,3),
     1 PQLEG(90,12),LCS(90,10),ILEG(1400),PN(90),QN(90),FEX(90)
      COMMON/SUB/PARM(150,9),PM(1500),QM(1500),P(300),Q(300),C(300)
     1,Z(300),RHO(20),S2ORHO(20),VISC(20),BULK(20),TEMP(20),PVAP(20)
     2,ATPRES,T,DELT,TFINAL,PLTDEL,PI,TITLE(20),LEGN,ICON
     3,KTEMP(99),LSTART(150),NLPT(150),LTYPE(99),NC(99),INX,INZ
     4,INV,ISTEP,NLINE,NEL,IND,IENTR,MNLINE,MNEL,MNLEG,MNNODE,MNPLOT
     5,MNLPTS,MDS
      DIMENSION D(6),DT(10),DD(1),L(7)
      INTEGER AREA1,AREA2,VOL1,VOL2,STROKE,QNET,EXPRES,P2,DUM
     1 ,A1,A2,CDA,CDB,ZN
C ******** D VARIABLES *************
      DATA AREA1/1/,AREA2/2/,VOL1/3/,VOL2/4/,STROKE/5/,INPOS/6/,
     1A1/7/,CDA/8/,A2/9/,CDB/10/
C ********* DT VARIABLES *************
      DATA QNET/1/,EXPRES/2/,MINIQV/3/,MAXIQV/4/,IQV/5/,NAREAR/6/,
     1 NZ1/7/,NGZ/8/,P2/9/,DUM/10/,K1/11/,K2/12/,ZN/13/
      IF(IENTR) 1000,2000,3000
 1000 CONTINUE
      IF(INEL.NE.0)GO TO 1500
      DT(IQV)=(D(INPOS)*D(AREA2)+D(VOL2))*2.0/DELT
      DT(DUM)=DT(IQV)*DELT/2.0
      DT(MINIQV)=D(VOL2)*2.0/DELT
      DT(MAXIQV)=D(STROKE)*D(AREA2)*2.0/DELT+DT(MINIQV)
      L(9)=0
      DT(QNET)=0.0
      DT(NAREAR)=D(AREA1)/D(AREA2)
      DT(EXPRES)=ATPRES*(D(AREA2)-D(AREA1))/D(AREA2)
      D(A1)=D(A1)**2*PI/4.
      D(A2)=D(A2)**2*PI/4.
      RO=RHO(KTEMP(IND))
      DT(K1)=2.*D(A1)**2*D(CDA)**2/RO
      DT(K2)=2.*D(A2)**2*D(CDB)**2/RO
      RETURN
C
C*****STEADY STATE SECTION*****
C
 1500 CONTINUE
C
C     M IS THE BOOTSTRAP NODE, N IS THE LOW PRESSURE NODE
      L(9)=L(9)+1
      LCS(INEL,7)=5
      QA=PQLEG(INEL,1)
      QS=PQLEG(INEL,2)
      WRITE(6,4004)IND,NC(IND),L(9),KNEL,INX,INEL
 4004 FORMAT(2X,4H64-I,2X,6I3)
      IF(KNEL.GT.5) GO TO 1950
```

6.63.7 (Continued)

```
      IF(KNEL.NE.1) GO TO 1600
      M=LCS(INEL,3)
      IF(INX.NE.1) GO TO 1800
      M=LCS(INEL,2)
      GO TO 1800
 1600 QR=QA*QS
      N=LCS(INEL,3)
      IF(INX.NE.1) GO TO 1700
      QR=-QR
      N=LCS(INEL,2)
 1700 DT(QNET)=DT(QNET)+QR
 1800 IF(L(9).NE.NC(IND)-L(8)) RETURN
      IF(N.EQ.M) WRITE(6,1900)
      L(9)=0
      DT(NZ1)=DT(QNET)
      IF(PN(M).EQ.0.0) PN(M)=3000.
      IF(PN(N).EQ.0.0) PN(N)=PN(M)*DT(NAREAR)+DT(EXPRES)
      PMN=PN(M)*DT(NAREAR)+DT(EXPRES)
      QN(N)=((PMN+PN(N))*20.)/2.-DT(QNET)
      QN(M)=DT(QNET)*DT(NAREAR)
      DT(P2)=PN(N)
      PEX(N)=20.
      DT(QNET)=0.0
 1900 FORMAT(5X,45HRSVR62 REQUIRES TWO NODES FOR BOOTSTRAP FLOW    )
      RETURN
 1950 IF(KNEL-6)1999,1955,1960
 1955 D1=1./DT(K1)
      GO TO 1965
 1960 D1=1./DT(K2)
 1965 PQLEG(INEL,8)=PQLEG(INEL,8)+D1
      PQLEG(INEL,11)=PQLEG(INEL,11)-D1*QA**2*QS
      GO TO 1800
 1999 STOP 6401
C ******** STOP 6401 INDICATES KNEL IS LESS THAN 6 ******************
 2000 CONTINUE
      DT(P2)=P(L(2))
      DT(QNET)=DT(NZ1)
      DT(NZ1)=Z(L(1))*DT(NAREAR)**2
      DT(NGZ)=1.0/DT(NZ1)
      DT(ZN)=0.0
      NCI=NC(IND)-2
      DO 2100 I=2,NCI
      DT(NGZ)=DT(NGZ)+1./Z(L(1))
 2100 DT(ZN)=DT(ZN)+1./Z(L(I))
      RETURN
 3000 CONTINUE
      NC1=NC(IND)-2
      ITEST=0
      L1=L(1)
      L2=L(2)
```

6.63.7 (Continued)

```
      L3=L(3)
      L4=L(4)
      L5=L(5)
      L6=L(6)
      L7=L(7)
      CN2=0.0
      ITER=0
      R1=1./DT(K1)
      B3=Z(L6)
      R2=1./DT(K2)
      B4=Z(L7)
      DO 3050 I=2,NCI
 3050 CN2=CN2+C(L(I))/Z(L(I))
      Q1=Q(L1)
 3100 Q2=CN2-((C(L1)-Q1*Z(L1))*DT(NAREAR)+DT(EXPRES))*DT(ZN)
      Q2=Q2*DT(NAREAR)
      CL1=C(L1)-Q1*Z(L1)
      D3=ABS(C(L6)-CL1)
      Q6=((-B3+SQRT(B3**2+4.*R1*D3))/(2.*R1))*SIGN(1.,C(L6)-CL1)
      D4=ABS(C(L7)-CL1)
      Q7=((-B4+SQRT(B4**2+4.*R2*D4))/(2.*R2))*SIGN(1.0,(C(L7)-CL1))
      QTOT=Q2+Q6+Q7
      IF(ABS(Q1+QTOT).LT..001) GO TO 3200
      Q1=(Q1-QTOT)/2.
      ITER=ITER+1
      IF(ITER.GT.25) GO TO 3200
      GO TO 3100
 3200 Q(L1)=Q1
      Q(L6)=Q6
      Q(L7)=Q7
      P(L1)=C(L1)-Q(L1)*Z(L1)
      P(L6)=C(L6)-Q(L6)*Z(L6)
      P(L7)=C(L7)-Q(L7)*Z(L7)
      IF(ITEST.EQ.1)GO TO 3500
      DT(P2)=P(L1)*DT(NAREAR)+DT(EXPRES)
      DT(IQV)=DT(IQV)+DT(QNET)+Q2/DT(NAREAR)
      CALL XLIMIT(DT(IQV),Q2,CN2,DT(MINIQV),DT(MAXIQV))
      IF(CN2.EQ.0.0.AND.Q2/DT(NAREAR).NE.0.0)GO TO 3500
      CN2=0.0
      ZN=0.0
      DO 3400 I=2,NCI
      LI=L(I)
      ZN=ZN+1.0/Z(LI)
 3400 CN2=CN2+C(LI)/Z(LI)
      DT(P2)=CN2/ZN
      ITEST=1
      GO TO 3100
 3500 CONTINUE
      DO 3600 I=2,NCI
      LI=L(I)
```

6.63.7 (Continued)

```
      P(LI)=DT(P2)
 3600 Q(LI)=(C(LI)-P(LI))/Z(LI)
      DT(QNET)=QZ/DT(NAREAR)
      DT(DUM)=DT(IQV)*DELT/2.
      RETURN
      END
```

APPENDIX J

TYPE #108 DUAL SYSTEM, TANDEM VALVE CONTROLLED ACTUATOR

HYTRAN USER MANUAL (AFAPL-TR-76-43, VOL. I)

6.108 TYPE #108 DUAL SYSTEM, TANDEM VALVE CONTROLLED ACTUATOR

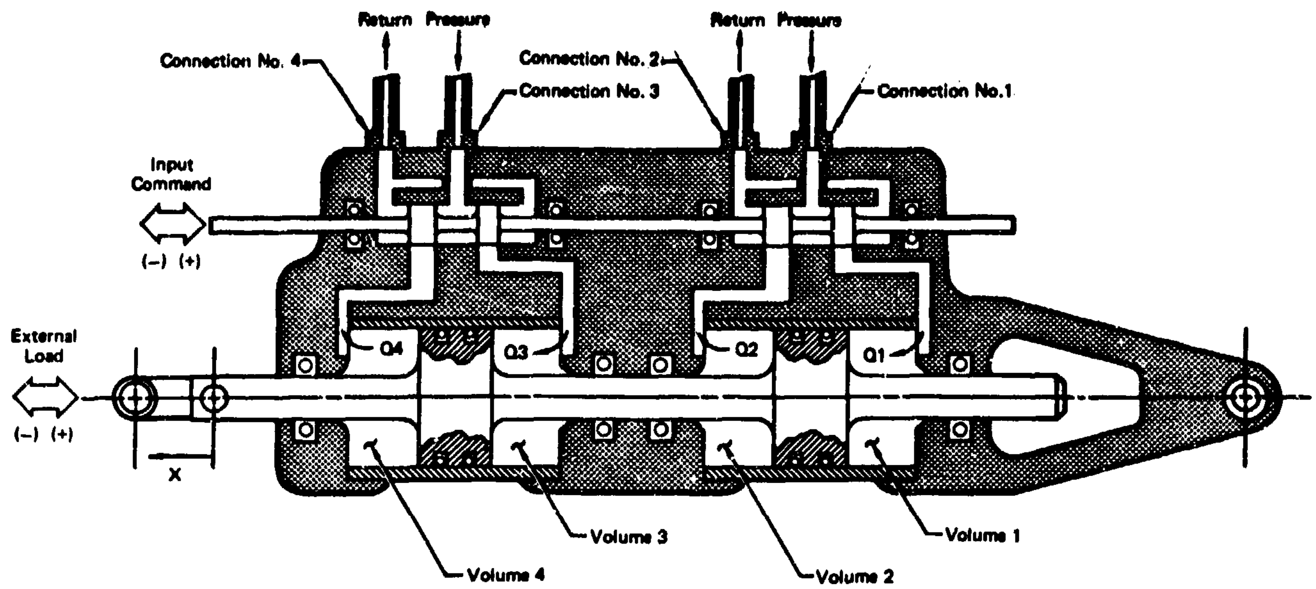


FIGURE 6.108-1
TYPE NO. 108 DUAL SYSTEM, TANDEM, VALVE CONTROLLED ACTUATOR

The dual system, tandem or parallel, valve controlled actuator has widespread use in aircraft flight control systems. The component is essentially two separate actuators whose valves move in unison and whose rams react on a common load. It is a straight forward method of obtaining survivability of a critical function. The subroutine provides a simple model of this actuator.

The model must begin a simulation with the valve closed (i.e. at time T = 0.0, valve position = 0.0). Positive valve displacement causes the actuator to extend, while negative valve displacement causes retraction. There is no feedback between the ram and the valve.

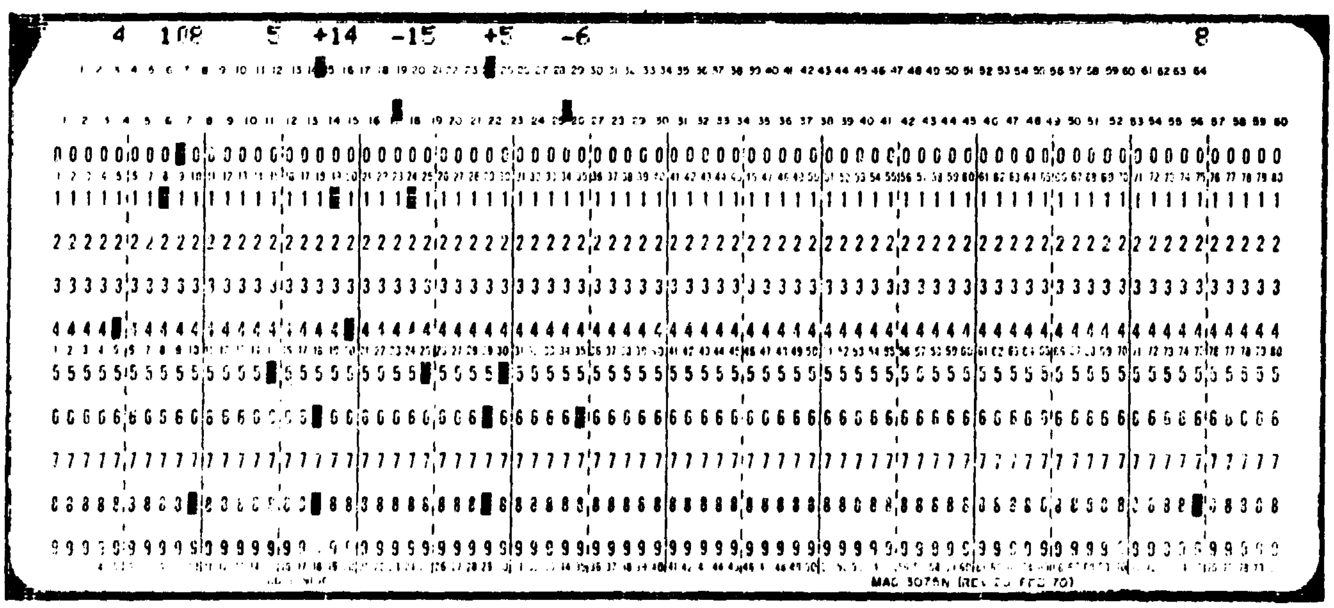
There are four connections on this component, and they must be made according to Figure 6.108-1. For either system, the incoming leg (pressure) should be numerically lower than the outgoing (return) leg. Nodes must be assigned to each piston of the actuator.

The subroutine includes cavitation simulations for both return connections.

CARD NUMBER 1

COLUMN	FORMAT	DATA
1-5	I5	Component Number
6-10	I5	Type Number = 108
11-15	I5	Number of Real Data Cards = 5 or more
16-20	I5	Line Number (with sign) attached to Connection 1
21-25	I5	Line Number (with sign) attached to Connection 2
26-30	I5	Line Number (with sign) attached to Connection 3
31-35	I5	Line Number (with sign) attached to Connection 4
36-40	I5	
41-45	I5	
46-50	I5	
51-55	I5	
56-60	I5	
61-65	I5	
66-70	I5	
71-75	I5	Number of data points in Time Table
76-80	I5	Temperature/Pressure Code (See Page 4.0-2)

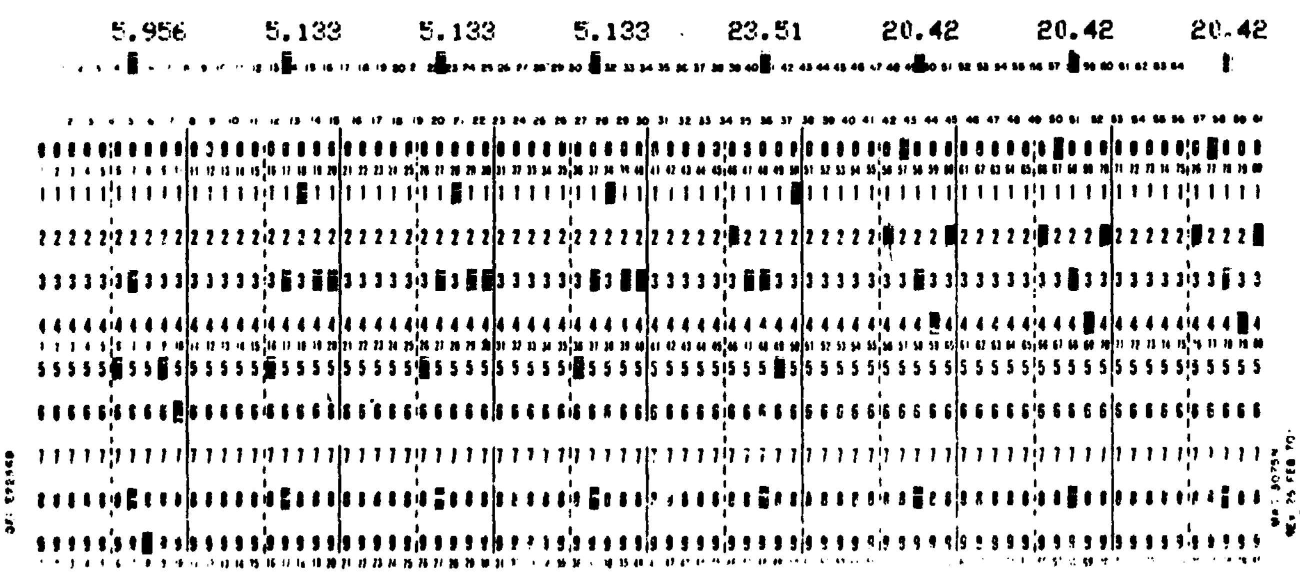
EXAMPLE CARD



CARD NUMBER 2

COLUMN	FORMAT	DATA	DIMENSIONS
1-10	E10.0	PISTON AREA #1	IN^2
11-20	E10.0	PISTON AREA #2	IN^2
21-30	E10.0	PISTON AREA #3	IN^2
31-40	E10.0	PISTON AREA #4	IN^2
41-50	E10.0	#1 VOLUME AT ZERO STROKE	IN^3
51-60	E10.0	#2 VOLUME AT ZERO STROKE	IN^3
61-70	E10.0	#3 VOLUME AT ZERO STROKE	IN^3
71-80	E10.0	#4 VOLUME AT ZERO STROKE	IN^3

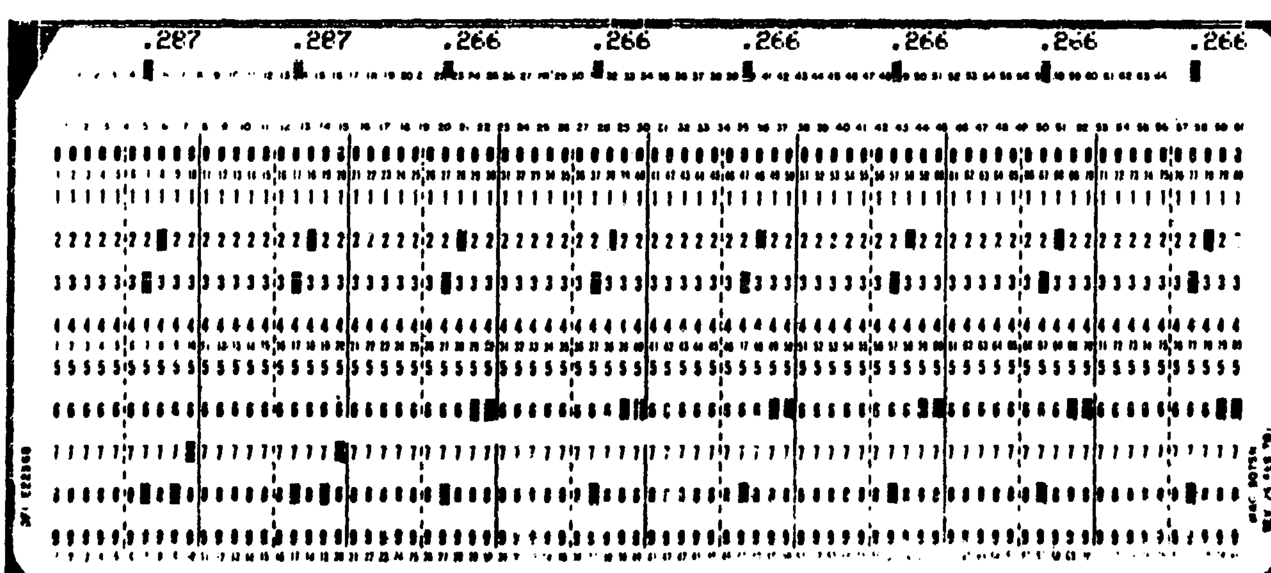
EXAMPLE CARD



CARD NUMBER 3

COLUMN	FORMAT	DATA	DIMENSIONS
1-10	E10.0	SLOT WIDTH: CON #1 TO VOL #1	IN
11-20	E10.0	SLOT WIDTH: CON #2 TO VOL #1	IN
21-30	E10.0	SLOT WIDTH: CON #1 TO VOL #2	IN
31-40	E10.0	SLOT WIDTH: CON #2 TO VOL #2	IN
41-50	E10.0	SLOT WIDTH: CON #3 TO VOL #3	IN
51-60	E10.0	SLOT WIDTH: CON #4 TO VOL #3	IN
61-70	E10.0	SLOT WIDTH: CON #3 TO VOL #4	IN
71-80	E10.0	SLOT WIDTH: CON #4 TO VOL #4	IN

EXAMPLE CARD

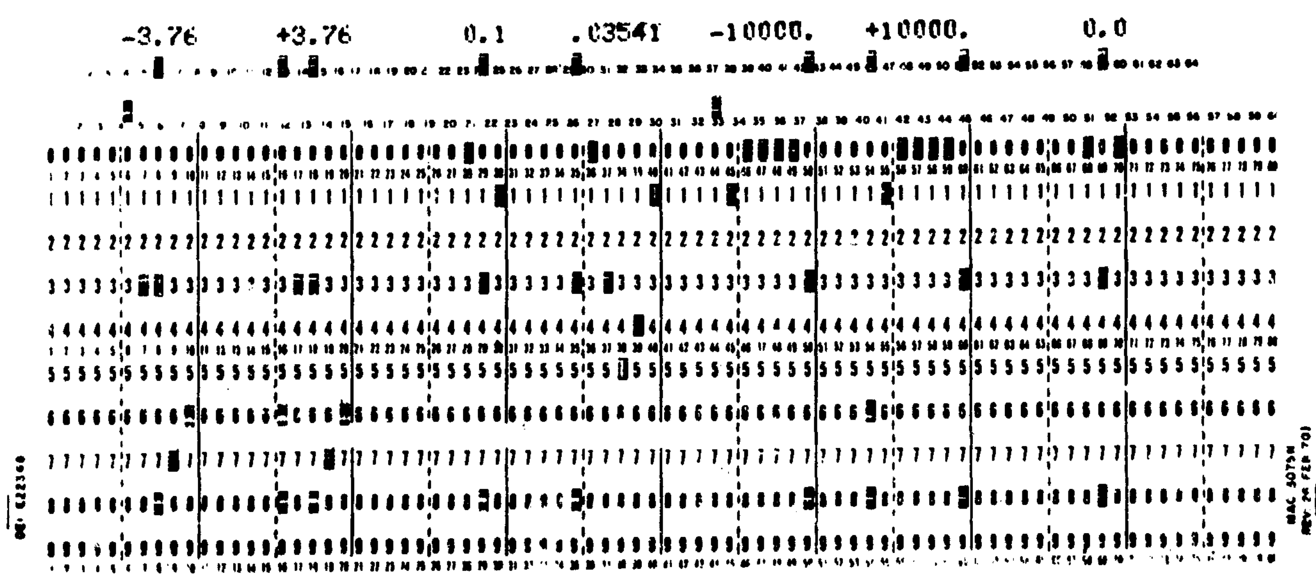


CARD NUMBER 4

COLUMN	FORMAT	DATA	DIMENSIONS
1-10	E10.0	STROKE FROM ZERO TO FULLY RETRACTED (0 OR -)	IN
11-20	E10.0	STROKE FROM ZERO TO FULLY EXTENDED (0 OR +)	IN
21-30	E10.0	VELOCITY DAMPING FACTOR	LBS*SEC/IN
31-40	E10.0	LOAD MASS (INCLUDING PISTONS + ROD)	LBS*SEC^2/IN
41-50	E10.0	LOAD AT FULLY RETRACTED POSITION*	LBS.
51-60	E10.0	LOAD AT FULLY EXTENDED POSITION*	LBS.
61-70	E10.0	INITIAL RAM POSITION	IN.
71-80	E10.0		

* COMPRESSIVE LOADS HAVE POSITIVE SIGN
TENSILE LOADS HAVE NEGATIVE SIGN

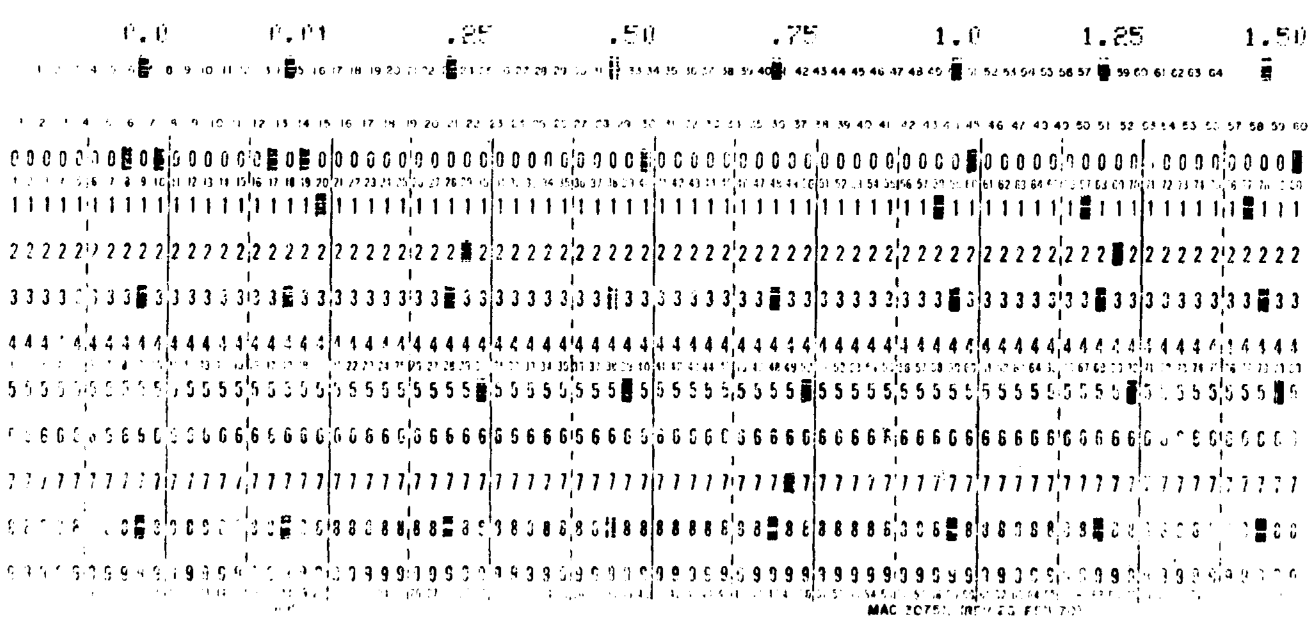
EXAMPLE CARD



CARD NUMBER 5

COLUMN	FORMAT	DATA	DIMENSIONS
1-10	E10.0	FIRST TIME VALUE (MUST BE 0.0)	SEC
11-20	E10.0	(ENTER AS MANY TIME VALUES AS	SEC
21-30	E10.0	NECESSARY USING AS MANY CARDS AS	SEC
31-40	E10.0	NECESSARY. THE TOTAL NUMBER SHOULD BE	SEC
41-50	E10.0	THE SAME AS THAT SPECIFIED ON CARD 1	SEC
51-60	E10.0	COL. 71-75, AND THE LAST TIME VALUE	SEC
61-70	E10.0	SHOULD BE GREATER THAN OR EQUAL TO THE	SEC
71-80	E10.0	FINAL CALCULATION TIME OF THE RUN.)	SEC

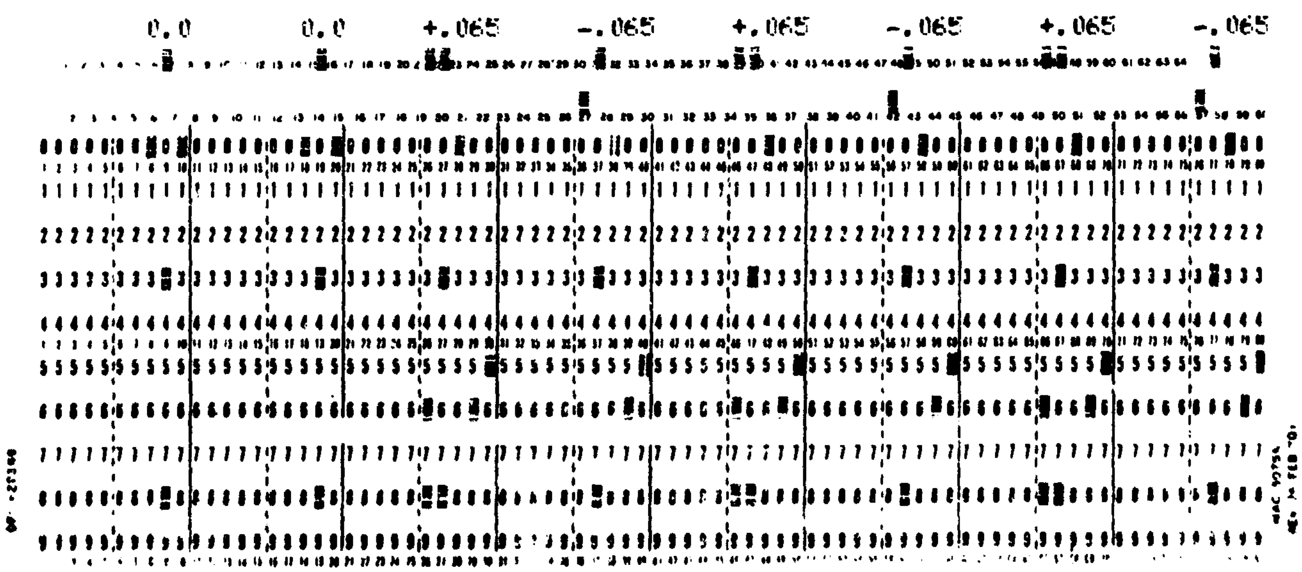
EXAMPLE CARD



CARD NUMBER 6

COLUMN	FORMAT	DATA	DIMENSIONS
1-10	E10.0	VALVE POSITION AT TIME T = 0.0 (MUST BE 0.0)	IN
11-20	E10.0	(ENTER AS MANY VALVE POSITION AS THERE	IN
21-30	E10.0	ARE TIME VALUES)	IN
31-40	E10.0		IN
41-50	E10.0		IN
51-60	E10.0		IN
61-70	E10.0		IN
71-80	E10.0		IN

EXAMPLE CARD



6.108 SUBROUTINE ACT108

The ACT108 subroutine models a tandem or parallel valve controlled actuator. Valve motion is input as a position/time history and the valve operates open loop without feedback.

Subroutine ACT108 can be used as a dual system actuator, but must begin the simulation with zero velocity (i.e. at time T = 0, valve position = 0) in order to work.

The valve is assumed to be of square port configuration with zero lap. The width of each port is input independently to allow the valve areas to be matched with their corresponding piston areas. Straight line interpolation is used to determine intermediate valve positons of the position/time table.

External loads at the fully retracted and fully extended positions may be input, and are assumed to vary linearly between these points. The effect of atmospheric pressure on an unbalanced area actuator is included.

The model is very similar to the simple valve controlled actuator modeled by subroutine ACT101.

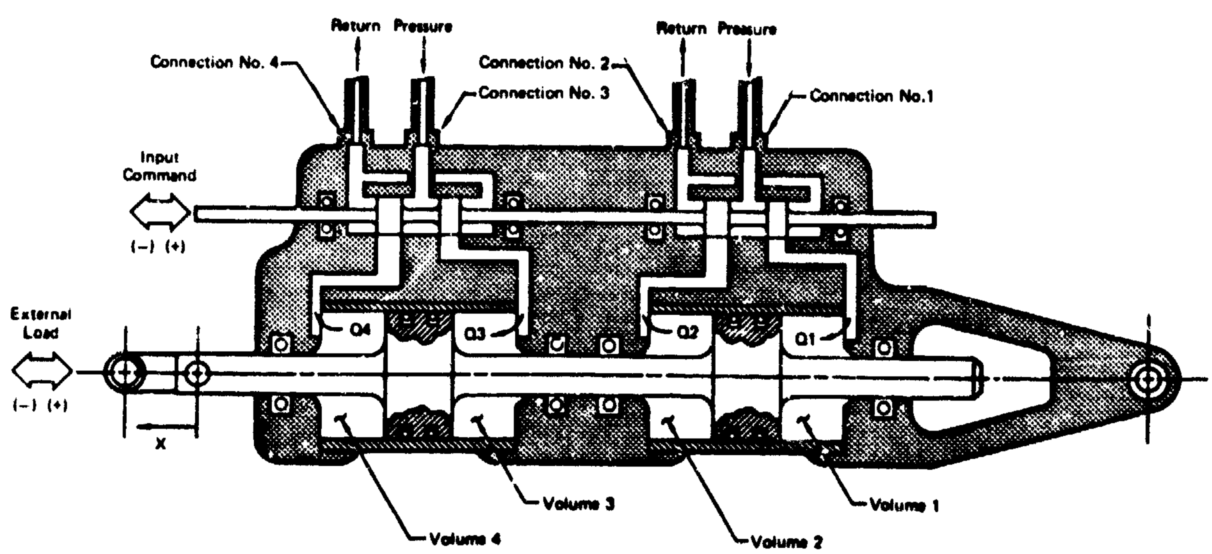


FIGURE 6.108-1
TYPE NO. 108 DUAL SYSTEM, TANDEM, VALVE CONTROLLED ACTUATOR

6.108.1 MATH MODEL

The math model for this subroutine can be divided into two parts, the flow calculations and the force balance.

The flow calculations are performed first, using the line equations orifice equations, volumetric impedance and predicted chamber pressures. The predicted chamber pressures are obtained from the actuator force balance performed during the previous time step.

For any given chamber of the actuator, by assigning a positive sign convention for flow into the chamber, the flow can be calculated using the general orifice formula:

FLOW = (SQRT(FN2**2+4 * FN1*ABS (C-PT))-FN2)/2.0 (1)

Where FN1 = (XV*VK)**2 (CIS^2/PSI)

FN2 = FN1*Z (CIS^2)

XV = valve position (IN)

VK = W*SQRT(2/Rho)*.65 $(CIS/PSI)^{1/2}$

Z = Line characteristic impedance + volumetric impedance (PSI/CIS)

C = Line characteristic pressure (PSI)

PT = Chamber volumetric pressure (PSI)

W = Valve slot width (IN)

Rho = Fluid density $(LB*SEC^2/IN^4)$

Use of the chamber volumetric impedance:

ZV = (Bulk Mod * Δt)/Volume

Where Bulk Mod = Bulk modulus of the fluid (PSI)

Volume = Chamber fluid volume (IN^4)

Δt = Time step increment (SEC)

Allows calculation of the chamber volumetric pressure as

PT = PP - VEL*AREA*ZV

Where PP = Predicted chamber pressure (PSI)

VEL = Actuator velocity (IN/SEC)

$AREA$ = Piston Area (IN^2)

The force balance on the actuator, as illustrated by Figure 6.108-2 is defined by the equation.

$$F_L = P1*A1-P2*A2+P3*A3-P4*A4-F_I-F_D \qquad (2)$$

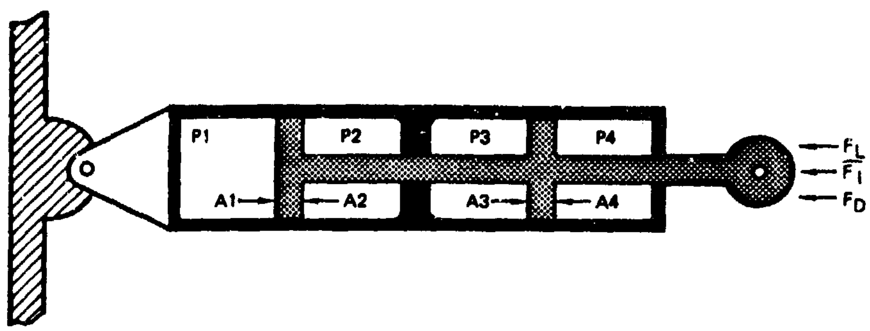


FIGURE 6.108-2

Where F_L = External Force (LB)

P1 = Chamber #1 Pressure (PSI)

A1 = Chamber #1 Piston Area (IN^2)

P2 = Chamber #2 Pressure (PSI)

A2 = Chamber #2 Piston Area (IN^2)

P3 = Chamber #3 Pressure (PSI)

A3 = Chamber #3 Piston Area (IN^2)

P4 = Chamber #4 Pressure (PSI)

A4 = Chamber #4 Piston Area (IN^2)

F_I = Inertia Force (LB)

F_D = Damping Force (LB)

Using the definitions

$$F_I = M*a = M*(Vn-Vo)/\Delta t$$

$$F_D = Vn*DAMP$$

Where M = Piston Mass ($LB/SEC^2/IN$)

a = Piston Acceleration (IN/SEC^2)

Vn = Piston Velocity (IN/SEC)

Vo = Last Velocity Calculation (IN/SEC)

Δt = Time Increment (SEC)

DAMP = Damping Factor (LB*SEC/IN)

The force balance (Equation 2) can be expressed as

$$P1*A1+P3*A3-P2*A2-P4*A4-V\ (M/\Delta t+DAMP)+Vo*M/\Delta t-F_L = 0 \qquad (3)$$

The actuator velocity is directly related to the actuator flows plus the chamber volume effects. Assigning positive displacement and velocity vectors to the actuator extend condition, the velocity can be related to the flows through the equations:

$$Vn*A1 = Q1 + (P1-P10)*GV1 \qquad (4)$$

$$-Vn*A2 = Q2 - (P2-P20)*GV2 \qquad (5)$$

$$Vn*A3 = Q3 * (P3-P30)*GV3 \qquad (6)$$

$$-Vn*A4 = Q4 - (P4-P40)*GV4 \qquad (7)$$

Where Q1 = Flow into Chamber #1 (CIS)

Q2 = Flow into Chamber #2 (CIS)

Q3 = Flow into Chamber #3 (CIS)

Q4 = Flow into Chamber #4 (CIS)

P10 = Previous time step's calculation of P1 (PSI)

P20 = Previous time step's calculation of P2 (PSI)

P30 = Previous time step's calculation of P3 (PSI)

P40 = Previous time step's calculation of P4 (PSI)

GV1 = Oil Volume #1 conductance (CIS/PSI)

GV2 = Oil Volume #2 conductance (CIS/PSI)

GV3 = Oil Volume #3 conductance (CIS/PSI)

GV4 = Oil Volume #4 Conductance (CIS/PSI)

During any time step, the oil volume conductance of any chamber, X , is calculated as

$$GV\ (X) = Volume\ (X) /\ BULK\ MOD * \Delta t$$

Rewriting Equations (4) through (7) in terms of P1, P2, P3, and P4 then substituting these into the force balance (Equation (3)), the actuator velocity can be solved for as

$$Vn = \frac{\frac{Q1*A1}{GV1} + \frac{Q3*A3}{GV3} - \frac{Q2*A2}{GV2} - \frac{Q4*A4}{GV4} + P10*A1+P30*A3-P20*A2-P40*At*V**INERT - F}{\frac{A1^2}{GV1} + \frac{A3^2}{GV3} + \frac{A2^2}{GV2} + \frac{A4^2}{GV4} + DAMP} \qquad (8)$$

The velocity may then be used to update the predicted chamber pressures through the equations

PP1 = PP1 + (Q1-Vn*A1)/GV1

PP2 = PP2 + (Q2+Vn*A2)/GV2

PP3 = PP3 + (Q3-Vn*A3)/GV3

PP4 = PP4 + (Q4+Vn*A4/GV4

The predicted chamber pressures and the newly calculated velocity are now ready for use in the next time step's calculations.

6.108.2 ASSUMPTIONS

In order to make this a generalized model, several important actuator characteristics have been omitted. These include cross piston leakage, manifold pressure losses, seal friction, barrel expansion of the actuator, external load spring mass characteristics, and valve leakage and neutral gain characteristics. All of these factors are highly specific to a given actuator design, and to include them would result in a model too detailed for general usage.

The model uses predicted values of several parameters in its solution technique. The assumption is made that these values are correct, when in fact there is some degree of error. The error decreases with decreasing calculation time intervals. Inaccuracies and instabilities can result if too large a time step is chosen for the simulation.

The ACT108 subroutine provides a simple, generalized model of what can be a very complex piece of hardware. The user must bear in mind that inaccuracies of some degree will result in any simulation because of the assumptions listed above. Judicious selection of input data can minimize the errors resulting from these assumptions.

6.108.3 COMPUTATION

SECTION 1000

Orifice characteristics for each flow path are calculated using the density of the fluid at the component, an assumed valve discharge coefficient of .65, and the valve slot widths input on the data cards.

These values are stored in the D array for use in the later sections of the program.

The slope of the load/stroke relationship is calculated and stored for use in the transient load calculations later on. This variable is used to determine the external load at the initial actuator position specified on the data cards. The load, along with atmospheric effects on an unbalanced area actuator, will be necessary for the steady state initialization performed in Section 1500.

Damping, bulk modulus, and inertia constants are calculated from the input data and stored as DT variables for later use.

The starting point of the position/time "table" for the valve is established, and the counters and variables necessary for the steady state calculations are initialized before control is returned to the main program.

SECTION 1500

The steady state portion of this subroutine assumes the valve is closed at the start of the simulation. To begin a simulation with an open valve (i.e. a moving actuator) would involve a solution technique not yet devised. This is because a tandem actuator is an interface point between two separate hydraulic systems. No inter-system flow exists, but there is a definite mechanical connection between the systems. A steady state balance on such a device would involve an integral force balance approach.

The first action in the 1500 section is to determine the initial position of the actuator ram with respect to its stroke limits. If the ram is somewhere between its stops, a force balance (performed separately for each piston) must be done to relate the chamber pressures. If the ram is at either one of its stops, the actuator can be in a bottomed condition, in which case the chamber pressures need not result in a force balance. For the bottomed condition, the piston leakage coefficient is set equal to 40000 PSI/CIS to allow some flow through the component for pressure calculations.

If the ram is not bottomed, the pressure calculations are done with a separate system force balance and no leakage impedance is necessary.

With the ram position determined, the next test is a check for which connection of the actuator the leg under calculation is attached to. If the test shows either connections 1 or 3 (i.e. the "pressure" ports of the actuator) to be in the leg, the assumption that each piston reacts one half of the external load is used to calculate the variable DT (PFORCE) as;

$$DT\ (PFORCE) = \left[PND * (EXTA-RETA) - EXTL/2. \right] /RETA$$

Where: PND = Downstream Node Pressure (PSI)

EXTA = Piston Extend Area (IN^2)

RETA = Piston Retract Area (IN^2)

EXTL = External Load (LB)

After this calculation, the leg constant for the Q^2 term is updated with the addition of 400000 PSI/CIS^2 and the chamber pressure set equal to the downstream node pressure before control is returned.

If connection 2 or 4 is in the leg, 400000 PSI/CIS^2 is added to the Q^2 leg term, the Q term is updated, and DT (PFORCE) is added to the constant pressure drop term. The upstream node pressure and chamber pressure are then calculated using DT (PFORCE), the leakage factors, and leg flow before control is again returned to the main program.

SECTION 2000

The predicted chamber pressures, which will be necessary for the first transient calculations, are set equal to the chamber pressures calculated in the steady state section. DT variables for the 3000 section are initialized, then control is returned to the calling program.

SECTION 3000

The oil volume conductance is calculated for each chamber based on the predicted chamber pressures and predicted ram position from the previous time step.

A call to the INTERP subroutine determines the valve position for the time step being calculated. If the valve is closed, all line and chamber flows are set equal to zero and control is routed to the line pressure equations. If the valve has a positive (actuator extend) or negative (actuator retract) displacement, tests determine which valve slot characteristics to use in the flow calculation given by Equation (1). The direction of this absolute value flow is determined by comparing the line characteristic pressure and predicted chamber pressure.

After all four line flows are calculated, the basic line equations are used to give the port pressures. Checks are then made to determine if either return port has cavitated. If this is so, the same type cavitation model as used in subroutine REST41 is used to assign chamber pressures and flows to track the resultant cavitation bubbles.

After the chamber flows are established, constants are calculated and the external load is predicted so the actuator ram velocity can be determined using Equation (8). Using this velocity, the new ram position is calculated and checked against the stroke limits before a position prediction is made for use in the next time step. The velocity, chamber flows, and oil impedances allow updating of the predicted chamber pressures for use in the next time step's calculations. Control is then returned to the main program.

6.108.4 LIMITATIONS

The major limiting factors of this subroutine are the need to begin the simulation with zero ram velocity and the straight line load/stroke relationship.

The desire for a generalized model has placed limitations on this subroutine when it is used to simulate a specific actuator design. The straight line valve characteristics may not give the same valve stroke/actuator response that an actual piece of hardware would exhibit because the valve slots are not always of rectangular configuration. Lack of manifold simulation and the assumption of a constant discharge coefficient will often give ram velocities that differ from what is observed on the actual hardware.

The subroutine is a good generalized model of a dual system actuator. It can be used to study waterhammer transients and inter-system performance, but the user should be aware of its limitations.

6.108.5 Variable Names

NAME	DESCRIPTION	DIMENSIONS
D(AREA1)	Piston Area #1 (EXT)	IN^2
D(AREA2)	Piston Area #2 (RET)	IN^2
D(AREA3)	Piston Area #3 (EXT)	IN^2
D(AREA4)	Piston Area #4 (RET)	IN^2
D(DAMP)	Damping Factor	LB*SEC/IN
D(INPOS)	Initial Piston Position	IN
D(MASS)	Piston/Rod Mass	$LB*SEC^2/IN$
D(MAXL)	External Load at Fully Extended Position	LB
D(MAXST)	Stroke from Zero Position to Fully Extended Position	IN
D(MINL)	External Load at Fully Retracted Position	LB
D(MINST)	Stroke from Zero Position to Fully Retracted Position	IN
D(SLOT11)	Valve Characteristic (Connection #1 to Chamber #1)	$IN^2/(\sqrt{PSI}*SEC)$
D(SLOT12)	Valve Characteristic (Connection #1 to Chamber #2)	$IN^2/(\sqrt{PSI}*SEC)$
D(SLOT21)	Valve Characteristic Connection #2 to Chamber #1)	$IN^2/(\sqrt{PSI}*SEC)$
D(SLOT22)	Valve Characteristic (Connection #2 to Chamber #2)	$IN^2/(\sqrt{PSI}*SEC)$
D(SLOT33)	Valve Characteristic (Connection #3 to Chamber #3)	$IN^2/(\sqrt{PSI}*SEC)$
D(SLOT34)	Valve Characteristic (Connection #3 to Chamber #4)	$IN^2/(\sqrt{PSI}*SEC)$
D(SLOT43)	Valve Characteristic (Connection #4 to Chamber#3)	$IN^2/(\sqrt{PSI}*SEC)$
D(SLOT44)	Valve Characteristic (Connection #4 to Chamber #4)	$IN^2/(\sqrt{PSI}*SEC)$
D(VOL1)	Chamber #1 Volume at Zero Stroke	IN^3
D(VOL2)	Chamber #2 Volume at Zero Stroke	IN^3
D(VOL3)	Chamber #3 Volume at Zero Stroke	IN^3

NAME	DESCRIPTION	DIMENSIONS
D(VOL4)	Chamber #4 Volume at Zero Stroke	IN^3
DELTO2	Simulation Time Step/2	SEC
DP1	Temporary Variable	PSI
DP2	Temporary Variable	PSI
DT(FORCE)	External Force at Initial Position	LB
DT(INERT)	Inertia Characteristic	LB*SEC/IN
DT(KBULK)	Bulk Modulus Factor	PSI/SEC
DT(KDAMP)	Inertia/Damping Factor	LB*SEC/IN
DT(LOADS)	Slope of Load/Stroke Relation	LB/IN
DT(LOADEX)	Atmospheric Force on Rod Due to Area Unbalance	LB
DT(LOADZ)	External Load at Zero Stroke	LB
DT(NCAV1)	Cavitation Monitor at Connection 2	CIS
DT(NCAV2)	Cavitation Monitor at Connection 4	CIS
DT(PFORCE)	Piston Force	LB
DT(PP1)	Pressure in Chamber #1	PSI
DT(PP2)	Pressure in Chamber #2	PSI
DT(PP3)	Pressure in Chamber #3	PSI
DT(PP4)	Pressure in Chamber #4	PSI
DT(PP1P)	Chamber #1 Predicted Pressure	PSI
DT(PP2P)	Chamber #2 Predicted Pressure	PSI
DT(PP3P)	Chamber #3 Predicted Pressure	PSI
DT(PP4P)	Chamber #4 Predicted Pressure	PSI
DT(PX)	Predicted Ram Position	IN
DT(Q1)	Chamber #1 Flow	CIS

NAME	DESCRIPTION	DIMENSIONS
DT(Q2)	Chamber #2 Flow	CIS
DT(Q3)	Chamber #3 Flow	CIS
DT(Q4)	Chamber #4 Flow	CIS
DT(VEL)	Actuator Ram Velocity	IN/SEC
DT(X)	Actuator Ram Position	IN
FLOW	Temporary Variable	CIS
FLOW1	Temporary Variable	CIS
FLOW2	Temporary Variable	CIS
FN1	Temporary Variable	CIS^2/PSI
FN2	Temporary Variable	CIS^2/PSI
FN3	Temporary Variable	CIS^2/PSI
FN4	Temporary Variable	CIS^2/PSI
GV1	Oil Volume #1 Conductance	CIS/PSI
GV2	Oil Volume #2 Conductance	CIS/PSI
GV3	Oil Volume #3 Conductance	CIS/PSI
GV4	Oil Volume #4 Conductance	CIS/PSI
QA2	Connection #2 Steady State Flow	CIS
QA4	Connection #4 Steady State Flow	CIS
QS2	Flow Sign	--
QS4	Flow Sign	--
VAP	Fluid Vapor Pressure	PSI
VK1	Temporary Variable	$IN^2/(\sqrt{PSI}*SEC)$
VK2	Temporary Variable	$IN^2/(\sqrt{PSI}*SEC)$

NAME	DESCRIPTION	DIMENSIONS
VK3	Temporary Variable	$IN^2/(\sqrt{PSI}*SEC)$
VK4	Temporary Variable	$IN^2/(\sqrt{PSI}*SEC)$
XV	Valve Position	IN
ZQ	Leakage Impedance	PSI/CIS
ZT1	Combined Line/Volume Impedance	PSI/CIS
ZT2	Combined Line/Volume Impedance	PSI/CIS
ZT3	Combined Line/Volume Impedance	PSI/CIS
ZT4	Combined Line/Volume Impedance	PSI/CIS
ZV1	Oil Volume #1 Impedance	PSI/CIS
ZV2	Oil Volume #2 Impedance	PSI/CIS
ZV3	Oil Volume #3 Impedance	PSI/CIS
ZV4	Oil Volume #4 Impedance	PSI/CIS

6.108.6 Subroutine Listing

```
      SUBROUTINE ACT108 (D,DT,DD,L)
C *** REVISED DECEMBER 1979 ***
C *** THIS SUBROUTINE MODELS A TANDEM VALVE CONTROLLED ACTUATOR
      DIMENSION D(32),DT(25),DD(1),L(14)
      COMMON NTELPL,NTOLPL,IPT,IPOINT,NPTS,INEL,KNEL,NTOPL,NLPLT(61,3),
     1 PQLEG(90,12),LCS(90,10),ILEG(1400),PN(90),QN(90)
      COMMON/SUB/PARM(150,9),PM(1500),QM(1500),P(300),Q(300),C(300)
     1,Z(300),RHO(20),S2ORHO(20),VISC(20),BULK(20),TEMP(20),PVAP(20)
     2,ATPRES,T,DELT,TFINAL,PLTDEL,PI,TITLE(20),LEGN,ICON
     3,KTEMP(99),LSTART(150),NLPT(150),LTYPE(99),NC(99),INX,INZ
     4,INV,ISTEP,NLINE,NEL,IND,IENTR,MNLINE,MNEL,MNLEG,MNNODE,MNPLOT
     5,MNLPTS,MDS
      INTEGER AREA1,AREA2,AREA3,AREA4,VOL1,VOL2,VOL3,VOL4,
     1SLOT11,SLOT12,SLOT21,SLOT22,SLOT33,SLOT34,SLOT43,SLOT44,
     2DAMP,X,VEL,PP1,PP2,PP3,PP4,PP1P,PP2P,PP3P,PP4P,PFORCE,
     3FORCE,CT,PX,Q1,Q2,Q3,Q4
C     D ARRAY VARIABLES
      DATA AREA1/1/,AREA2/2/,AREA3/3/,AREA4/4/,VOL1/5/,VOL2/6/,
     1VOL3/7/,VOL4/8/,SLOT11/9/,SLOT12/10/,SLOT21/11/,SLOT22/12/,
     2SLOT33/13/,SLOT34/14/,SLOT43/15/,SLOT44/16/,MINST/17/,MAXST/18/,
     3DAMP/19/,MASS/20/,MINL/21/,MAXL/22/,INPOS/23/
C     L ARRAY VARIABLES
      DATA NTAB/5/,IY/6/,NODE/7/,CT/8/,LG1/9/,LG2/10/,LG3/11/,LG4/12/,
     1 N1/13/,N2/14/
C     DT ARRAY VARIABLES
      DATA X/1/,PX/2/,VEL/3/,Q1/4/,Q2/5/,Q3/6/,Q4/7/,PP1/8/
     1,PP2/9/,PP3/10/,PP4/11/,PP1P/12/,PP2P/13/,PP3P/14/,
     2 PP4P/15/,FORCE/16/,PFORCE/17/,LOADS/18/,LOADZ/19/,
     3 LOADEX/20/,INERT/21/,KDAMP/22/,KBULK/23/,NCAV1/24/,NCAV2/25/
C
      IF(IENTR) 1000,2000,3000
C *** 1000 SECTION
 1000 CONTINUE
      IF(INEL.NE.0)GO TO 1500
C     ACTUATOR PARAMETER INPUT
      D(SLOT11)=D(SLOT11)*0.65*S2ORHO(KTEMP(IND))
      D(SLOT12)=D(SLOT12)*0.65*S2ORHO(KTEMP(IND))
      D(SLOT21)=D(SLOT21)*0.65*S2ORHO(KTEMP(IND))
      D(SLOT22)=D(SLOT22)*0.65*S2ORHO(KTEMP(IND))
      D(SLOT33)=D(SLOT33)*0.65*S2ORHO(KTEMP(IND))
      D(SLOT34)=D(SLOT34)*0.65*S2ORHO(KTEMP(IND))
      D(SLOT43)=D(SLOT43)*0.65*S2ORHO(KTEMP(IND))
      D(SLOT44)=D(SLOT44)*0.65*S2ORHO(KTEMP(IND))
      DT(LOADS)=(D(MAXL)-D(MINL))/(D(MAXST)-D(MINST))
      DT(LOADZ)=D(MAXL)-DT(LOADS)*D(MAXST)
      DT(FORCE)=DT(LOADZ)+DT(LOADS)*D(INPOS)
      DT(LOADEX)=ATPRES*(D(AREA1)-D(AREA2)+D(AREA3)-D(AREA4))
      DT(INERT)=D(MASS)/DELT
      DT(KDAMP)=D(DAMP)+DT(INERT)
      DT(X)=D(INPOS)
```

6.108.6 (Continued)

```
      DT(KBULK)=BULK(KTEMP(IND))*DELT
      DELTO2=DELT/2.0
      L(NTAB)=L(12)
      L(IY)=(L(NTAB)+7)/8
      L(IY)=25+L(IY)*8
      L(NODE)=1
      L(CT)=0
      XV=D(L(IY))
      DT(26)=XV
      DT(PP1P)=400000.
      DT(PP2P)=400000.
      DT(PP3P)=400000.
      DT(PP4P)=400000.
      DT(NCAV1)=0.0
      DT(NCAV2)=0.0
      RETURN
C
C *** 1500 SECTION
C     THE STEADY STATE SECTION
 1500 CONTINUE
      ZQ=0.0
      IF(DT(X).GT.D(MINST).AND.DT(X).LT.D(MAXST))GO TO 1505
      ZQ=40000.
 1505 GO TO (10,20,30,40),KNEL
   10 L(LG1)=INEL
      L(N1)=LCS(INEL,3)
      L(CT)=L(CT)+1
      DT(PFORCE)=(PN(L(N1))*(D(AREA1)-D(AREA2))-DT(FORCE)/2.)/D(AREA2)
      PQLEG(INEL,8)=PQLEG(INEL,8)+DT(PP1P)
      DT(PP1)=PN(L(N1))
      GO TO 1550
   20 L(LG2)=INEL
      L(CT)=L(CT)+1
      QA2=PQLEG(INEL,1)
      QS2=PQLEG(INEL,2)
      PQLEG(INEL,8)=PQLEG(INEL,8)+DT(PP2P)
      PQLEG(INEL,6)=PQLEG(INEL,6)+ZQ
      PQLEG(INEL,5)=PQLEG(INEL,5)+DT(PFORCE)
      PQLEG(INEL,11)=PQLEG(INEL,11)+DT(PFORCE)-QA2*QS2*(ZQ+QA2*
     1DT(PP2P))
      DT(PP2)=PN(L(N1))+DT(PFORCE)-QA2*QS2*ZQ
      GO TO 1550
   30 L(LG3)=INEL
      L(N2)=LCS(INEL,3)
      L(CT)=L(CT)+1
      DT(PFORCE)=(PN(L(N2))*(D(AREA3)-D(AREA4))-DT(FORCE)/2.)/D(AREA4)
      PQLEG(INEL,8)=PQLEG(INEL,8)+DT(PP3P)
      DT(PP3)=PN(L(N2))
      GO TO 1550
   40 L(LG4)=INEL
```

6.108.6 (Continued)

```
      L(CT)=L(CT)+1
      QA4=PQLEG(INEL,1)
      QS4=PQLEG(INEL,2)
      PQLEG(INEL,8)=PQLEG(INEL,8)+DT(PP4P)
      PQLEG(INEL,6)=PQLEG(INEL,6)+ZQ
      PQLEG(INEL,5)=PQLEG(INEL,5)+DT(PFORCE)
      PQLEG(INEL,11)=PQLEG(INEL,11)+DT(PFORCE)-QA4*QS4*(ZQ+QA4*
     1DT(PP4P))
      DT(PP4)=PN(L(N2))+DT(PFORCE)-QA4*QS4*ZQ
 1550 IF(L(CT).EQ.4)GO TO 1600
      RETURN
 1600 CONTINUE
      WRITE(6,4100)L(LG1),L(LG2),L(LG3),L(LG4),L(N1),L(N2)
 4100 FORMAT(2X,3H108,6I3)
      L(CT)=0
      WRITE(6,4200)PN(L(N1)),PN(L(N2)),QA2,QA4,DT(PP1),DT(PP2)
     1,DT(PP3),DT(PP4),PQLEG(L(LG2),11),PQLEG(L(LG4),11)
      WRITE(6,4300)PQLEG(L(LG1),8),PQLEG(L(LG2),8),PQLEG(L(LG3),8)
     1,PQLEG(L(LG4),8),PQLEG(L(LG2),6),PQLEG(L(LG4),6)
     2,PQLEG(L(LG2),5),PQLEG(L(LG4),5)
 4200 FORMAT(2X,4H108A,10E12.4)
 4300 FORMAT(2X,4H108B,8E12.4)
      RETURN
 1900 WRITE(6,1950) IND,KNEL,INEL
 1950 FORMAT(5X,7HCOMP NO,I3,20H, HAS INVALID CON NO  ,I3,
     1 11H, IN LEG NO ,I4)
C *** 2000 SECTION
 2000 CONTINUE
      DT(PP1P)=DT(PP1)
      DT(PP2P)=DT(PP2)
      DT(PP3P)=DT(PP3)
      DT(PP4P)=DT(PP4)
      DT(PX)=DT(X)
      DT(VEL)=0.0
      DT(FORCE)=0.0
      XV=D(L(IY))
      IF(XV.GT.0.0)WRITE(6,2900)
 2900 FORMAT(2X,19HVALVE IS NOT AT 0.0)
      DT(Q2)=Q(L(2))
      DT(Q1)=Q(L(1))
      DT(Q3)=Q(L(3))
      DT(Q4)=Q(L(4))
      RETURN
C
C *** 3000 SECTION
 3000 CONTINUE
C
      L1=L(1)
      L2=L(2)
      L3=L(3)
```

6.108.6 (Continued)

```
      L4=L(4)
      GV1=(D(VOL1)+DT(PX)*D(AREA1))/DT(KBULK)
      ZV1=1.0/GV1
      DT(PP1P)=DT(PP1)-DT(VEL)*D(AREA1)*ZV1
      GV2=(D(VOL2)-DT(PX)*D(AREA2))/DT(KBULK)
      ZV2=1.0/GV2
      DT(PP2P)=DT(PP2)+DT(VEL)*D(AREA2)*ZV2
      GV3=(D(VOL3)+DT(PX)*D(AREA3))/DT(KBULK)
      ZV3=1.0/GV3
      DT(PP3P)=DT(PP3)-DT(VEL)*D(AREA3)*ZV3
      GV4=(D(VOL4)-DT(PX)*D(AREA4))/DT(KBULK)
      ZV4=1.0/GV4
      DT(PP4P)=DT(PP4)+DT(VEL)*D(AREA4)*ZV4
C
      CALL INTERP (T,D(25),D(L(IY)),10,L(NTAB),XV,IN2)
C
C     CALCULATE LINE FLOWS AND PRESSURES
      DT(26)=XV
      IF (XV) 140,170,180
C     XV LESS THAN 0
  140 VK1=D(SLOT21)
      VK2=D(SLOT43)
      ZT1=Z(L1)+ZV2
      ZT2=Z(L3)+ZV4
      DP1=C(L1)-DT(PP2P)
      DP2=C(L3)-DT(PP4P)
      ICALC=1
      GO TO 210
  150 DT(Q2)=SIGN(FLOW1,DP1)
      DT(Q4)=SIGN(FLOW2,DP2)
      Q(L1)=DT(Q2)
      Q(L3)=DT(Q4)
      VK1=D(SLOT12)
      VK2=D(SLOT34)
      ZT1=Z(L2)+ZV1
      ZT2=Z(L4)+ZV3
      DP1=C(L2)-DT(PP1P)
      DP2=C(L4)-DT(PP3P)
      ICALC=2
      GO TO 210
  160 DT(Q1)=SIGN(FLOW1,DP1)
      DT(Q3)=SIGN(FLOW2,DP2)
      Q(L2)=DT(Q1)
      Q(L4)=DT(Q3)
      GO TO 220
C
C     XV = 0
  170 DT(Q1)=0.
      DT(Q2)=0.
      DT(Q3)=0.0
```

6.108.6 (Continued)

```
      DT(Q4)=0.0
      Q(L2)=0.0
      Q(L1)=0.0
      Q(L3)=0.0
      Q(L4)=0.0
      ICALC=5
      GO TO 220
C
C     XV GREATER THAN 0
  180 VK1=D(SLOT11)
      VK2=D(SLOT33)
      ZT1=Z(L1)+ZV1
      ZT2=Z(L3)+ZV3
      DP1=C(L1)-DT(PP1P)
      DP2=C(L3)-DT(PP3P)
      ICALC=3
      GO TO 210
  190 DT(Q1)=SIGN(FLOW1,DP1)
      DT(Q3)=SIGN(FLOW2,DP2)
      Q(L1)=DT(Q1)
      Q(L3)=DT(Q3)
      VK1=D(SLOT22)
      VK2=D(SLOT44)
      ZT1=Z(L2)+ZV2
      ZT2=Z(L4)+ZV4
      DP1=C(L2)-DT(PP2P)
      DP2=C(L4)-DT(PP4P)
      ICALC=4
      GO TO 210
  200 DT(Q2)=SIGN(FLOW1,DP1)
      DT(Q4)=SIGN(FLOW2,DP2)
      Q(L2)=DT(Q2)
      Q(L4)=DT(Q4)
      GO TO 220
C
C     CALCULATE ABSOLUTE VALUE OF FLOWS
  210 FN1=XV*XV*VK1*VK1
      FN2=FN1*ZT1
      FN3=XV*XV*VK2*VK2
      FN4=FN3*ZT2
      FLOW1=(SQRT(FN2**2+4.0*FN1*ABS(DP1))-FN2)/2.0
      FLOW2=(SQRT(FN4**2+4.0*FN3*ABS(DP2))-FN4)/2.0
      GO TO (150,160,190,200),ICALC
C
  220 CONTINUE
      P(L1)=C(L1)-Q(L1)*Z(L1)
      P(L2)=C(L2)-Q(L2)*Z(L2)
      P(L3)=C(L3)-Q(L3)*Z(L3)
      P(L4)=C(L4)-Q(L4)*Z(L4)
      VAP=PVAP(KTEMP(IND))
```

6.108.6 (Continued)

```
      IF(P(L2).GT.VAP.AND.DT(NCAV1).LE.0.0) GO TO 300
      FLOW=0.0
      GO TO (230,230,240,240,250),ICALC
  230 FN2=FN1*ZV1
      DP=VAP-DT(PP1P)
      FLOW=(SQRT(FN2**2+4.0*FN1*ABS(DP))-FN2)/2.0
      FLOW=SIGN(FLOW,DP)
      DT(Q1)=FLOW
      GO TO 250
  240 FN2=FN1*ZV2
      DP=VAP-DT(PP2P)
      FLOW=(SQRT(FN2**2+4.0*FN1*ABS(DP))-FN2)/2.0
      FLOW=SIGN(FLOW,DP)
      DT(Q2)=FLOW
  250 Q(L2)=(C(L2)-VAP)/Z(L2)
      P(L2)=VAP
      DT(NCAV1)=DT(NCAV1)+FLOW-Q(L2)
      IF(DT(NCAV1).LT.0.0) DT(NCAV1)=0.0
  300 IF(P(L4).GT.VAP.AND.DT(NCAV2).LE.0.0)GO TO 400
      FLOW=0.0
      GO TO (330,330,340,340,350),ICALC
  330 FN4=FN3*ZV3
      DP=VAP-DT(PP3P)
      FLOW=(SQRT(FN4**2+4.0*FN3*ABS(DP))-FN4)/2.0
      FLOW=SIGN(FLOW,DP)
      DT(Q3)=FLOW
      GO TO 350
  340 FN4=FN3*ZV4
      DP=VAP-DT(PP4P)
      FLOW=(SQRT(FN4**2+4.0*FN3*ABS(DP))-FN4)/2.0
      FLOW=SIGN(FLOW,DP)
      DT(Q4)=FLOW
  350 Q(L4)=(C(L4)-VAP)/Z(L4)
      P(L4)=VAP
      DT(NCAV2)=DT(NCAV2)+FLOW-Q(L4)
      IF(DT(NCAV2).LT.0.0) DT(NCAV2)=0.0
  400 CONTINUE
C
C     CALCULATE ACTUATOR VELOCITY
      G1=GV1/D(AREA1)
      G2=GV2/D(AREA2)
      G3=GV3/D(AREA3)
      G4=GV4/D(AREA4)
      ZN=DT(KDAMP)+D(AREA1)/G1+D(AREA2)/G2+D(AREA3)/G3+D(AREA4)/G4
      DP=DT(LOADEX)+DT(LOADZ)+DT(PX)*DT(LOADS)
      DELTP=DT(Q1)/G1+DT(Q3)/G3+DT(PP1)*D(AREA1)+DT(PP3)*D(AREA3)
     1-DT(Q2)/G2-DT(Q4)/G4-DT(PP2)*D(AREA2)-DT(PP4)*D(AREA4)
      VELO=DT(VEL)
      DT(VEL)=(DELTP-DP+VELO*DT(INERT))/ZN
      DT(PX)=DT(X)
```

6.108.6 (Continued)

```
DT(X)=DT(X)+DELTO2*(DT(VEL)+VELO)
CALL XLIMIT(DT(X),DT(VEL),DP,D(MINST),D(MAXST))
DT(PX)=DT(X)*2.0-DT(PX)
IF(DP.NE.0.0) DT(PX)=DT(X)
IF(DP.EQ.0.0) VELO=DT(VEL)
DT(PP1)=DT(PP1)+(DT(Q1)-VELO*D(AREA1))/GV1
DT(PP2)=DT(PP2)+(DT(Q2)+VELO*D(AREA2))/GV2
DT(PP3)=DT(PP3)+(DT(Q3)-VELO*D(AREA3))/GV3
DT(PP4)=DT(PP4)+(DT(Q4)+VELO*D(AREA4))/GV4
RETURN
END
```

APPENDIX K

SYSTEM ARRANGEMENT DATA
HYTRAN USER MANUAL (AFAPL-TR-76-43, VOL. I)

7.0 SYSTEM ARRANGEMENT DATA

The system arrangement data is used to describe the system configuration to allow the steady state flow initialization necessary to begin the transient simulation. This input data consists of number coding that describes the flow paths in the system and identifies locations where the flow divides (such as a tee or cross) or where there is a flow disparity (such as an unbalanced area actuator).

NODE LOCATIONS

As a general "rule of thumb" every component in a system, except the following, will require at least one node to be placed at it.

- Two-way Control Valves (TYPE 21)
- Check Valves (TYPE 31)
- Priority Valves (TYPE 32)
- One-Way Restrictors (TYPE 33)
- Two-Way Restrictors (TYPE 41)
- Filters (TYPES 81, 82 & 83)

The frictionless branch (TYPE 11) requires a node to be placed at it whenever there is a division of flow at that location. The instances when a node is not required at a branch (TYPE 11) are:

- The branch has only two connections
- The branch has three or more connections but only two of them will experience steady state flow (the other connections serve as origin points for "zero flow" legs)

SPECIAL CASES

When a leg is terminated by a constant pressure source, that constant pressure must be input along with the leg connection information. Only nodes serving a single leg can be a constant pressure termination.

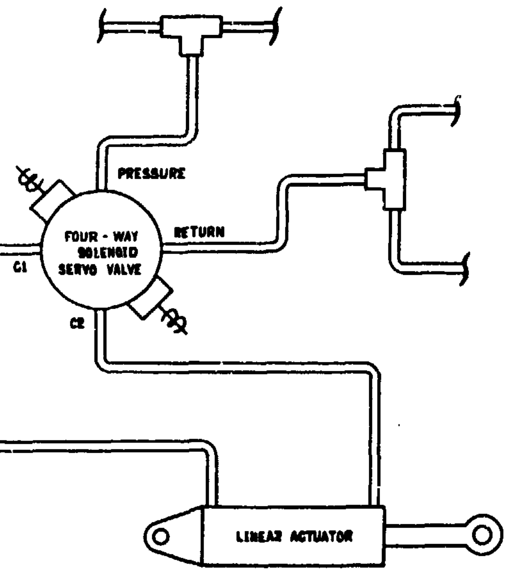
When using the constant pressure node capability, there must be at least one node in the circuit at which the pressure is not specified.

Nodes should not be placed in the center of any component having a pressure loss because each leg connected to the node will include the pressure drop of the component.

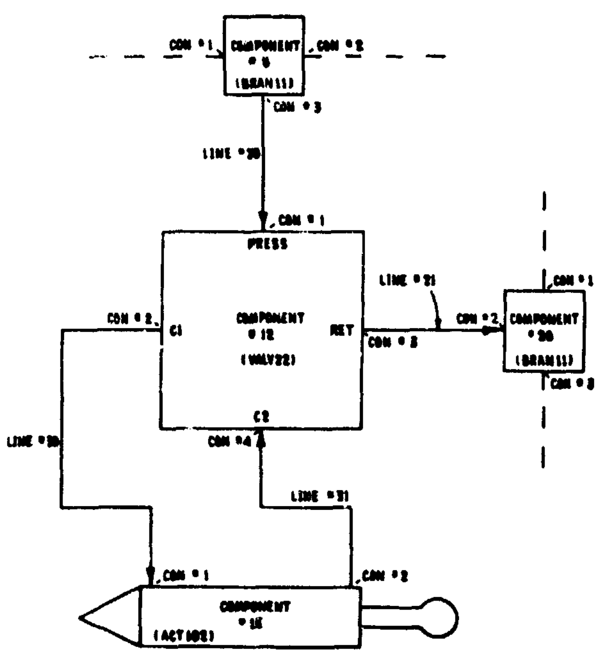
Certain components require special node /leg configurations. These components are:

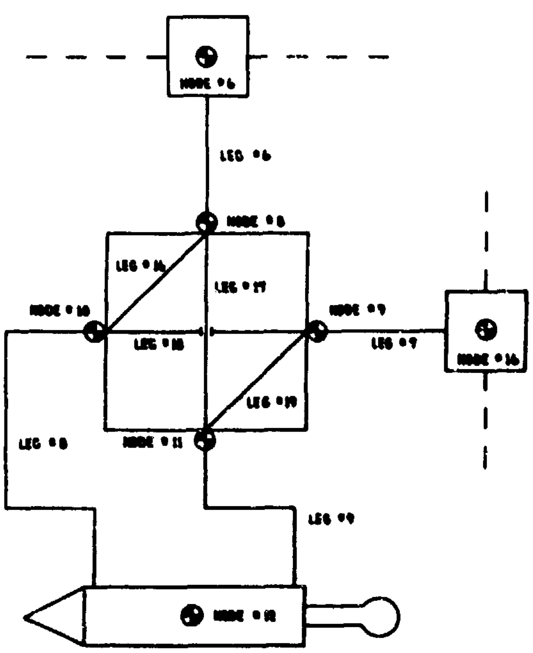
FOUR-WAY CONTROL VALVES (VALV22)

VALV22 requires 4 nodes for its steady state description. A node is placed at each port of the valve and is not considered internal to the valve. Legs that connect the valve to the rest of the system originate or terminate at the port nodes of the valve, but must not include the valve as a component of the leg. The internal legs of the valve connect the port nodes of the valve so as to describe the flow paths within the valve. When assembling the internal legs of the valve, it is important that each connection of the valve be called out as a component connection once, and only once. Figure is a schematic representation of a typical HYTRAN VALV22 usage. Figure is the steady state (leg & node) data necessary to describe this valve installation.



Actual Subsystem





HYTRAN Component & Line Description

HYTRAN Leg & Node Description

LEG #	UPSTREAM NODE #	DOWNSTREAM NODE #	# OF ELEMENTS IN LEG	FLOW GUESS (CIS)	
6	6	8	2	1.0	
8	3	0	20		
7	9	16	2	1.0	
0	21	20	2		
8	10	12	2	1.0	EXTERNAL LEGS
0	30	15	1		
9	12	11	2	1.0	
15	2	0	31		
16	8	10	1	1.0	
12	1				
17	8	11	1	1.0	
12	4				
18	10	9	1	1.0	INTERNAL VALVE LEGS
12	2				
19	11	9	1	1.0	
12	3				

SAMPLE LEG/NODE INPUT DATA FOR FIGURE EXAMPLE SUBSYSTEM

FIGURE

ACTUATORS

- ACT101 requires one node (located internally)
- ACT102 requires one node (located internally)
- ACT108 requires two nodes (one in each independent half)

RESERVOIRS

- RSVR61 requires one node which should not be set as a constant pressure node
- RSVR62 requires two open ended nodes (not connected by a leg). One node is considered to be in the high pressure (bootstrap) section, the other node is considered to be in the low pressure (reservoir) section.
- RSVR63 requires two nodes (open ended). One on the bootstrap/RLS circuit side, and one on the reservoir (low pressure) side.

7.1 GENERAL DATA

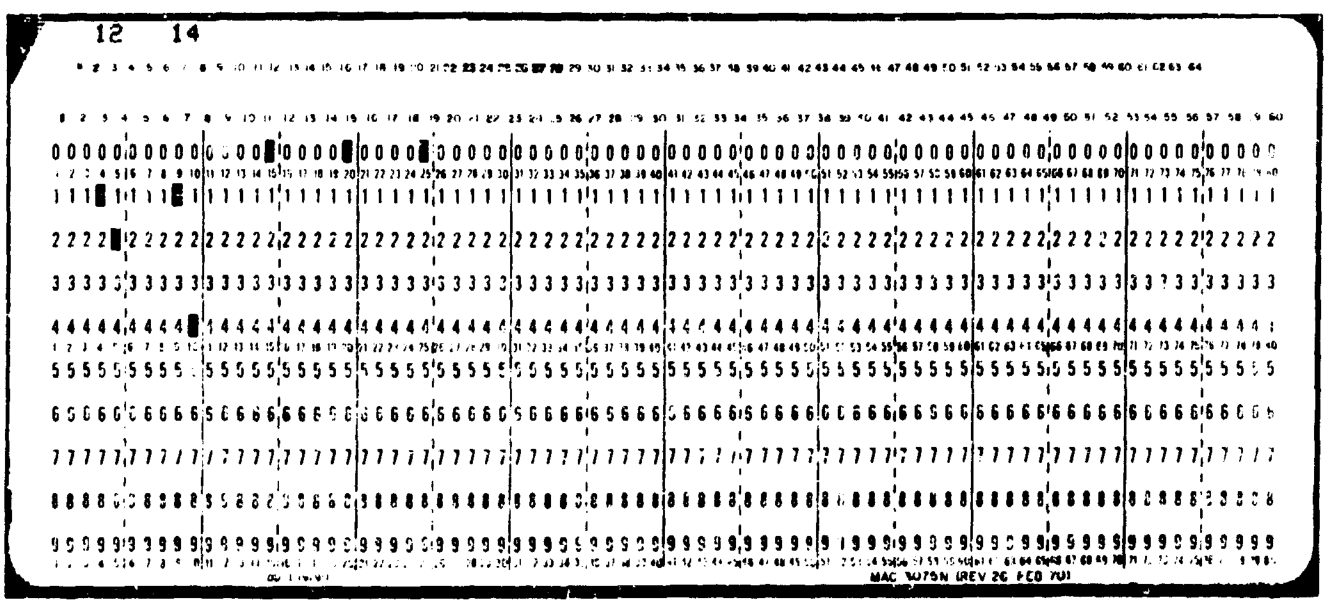
On this card input the number of nodes, the number of legs, the number of constant pressure points, the number of zero flow legs and the number of hydraulic systems.

A zero flow leg is a dead ended line with no steady state flow. The pressure at the end of the leg is determined by the steady state program.

GENERAL DATA CARD

COLUMN	FORMAT	DATA
1-5	I5	Number of Nodes
6-10	I5	Number of Legs
11-15	I5	Number of Constant Pressure Points
16-20	I5	Number of Zero Flow Legs
21-25	I5	Number of Systems
26-30	I5	
31-35	I5	
36-40	I5	
41-45	I5	
46-50	I5	
51-55	I5	
56-60	I5	
61-65	I5	
66-70	I5	
71-75	I5	
76-80	I5	

EXAMPLE CARD



7.2 LEG INPUT DATA

Two or more cards are required to input the data for each leg. The first card contains the leg number, upstream node number, downstream node number, number of elements in the leg, initial flow guess, constant pressure at upstream node if applicable and constant pressure at downstream node, if applicable.

The second card or cards contains the leg connection details. Starting with the component or line at the upstream node and progressing along the flow path to the downstream node, the element number and type are input. Because of the mixture of lines and components, the need to differentiate between the element numbering system is as follows:

First Pair of Data

First value	>0	Component number
	=0	Element is a line
Second value	=	*Component connection number or line number

*Use upstream connection if the component has upstream and downstream connections in the same leg.

This is repeated N times for the N elements in the leg.

Zero Flow Leg Data Cards

The zero flow leg data cards (if any) are entered after the leg input data cards. Two cards are required for each zero flow leg. The first card contains the zero flow leg number, the component number for the pressure reference on the zero flow leg and the number of elements in the zero flow leg. The upstream element must be a branch component and the zero flow leg should not include the branch as the first element.

The second data card contains the zero flow leg connection details starting with the line connected to the upstream branch. The data follows the sequence specified on Card Number 2.

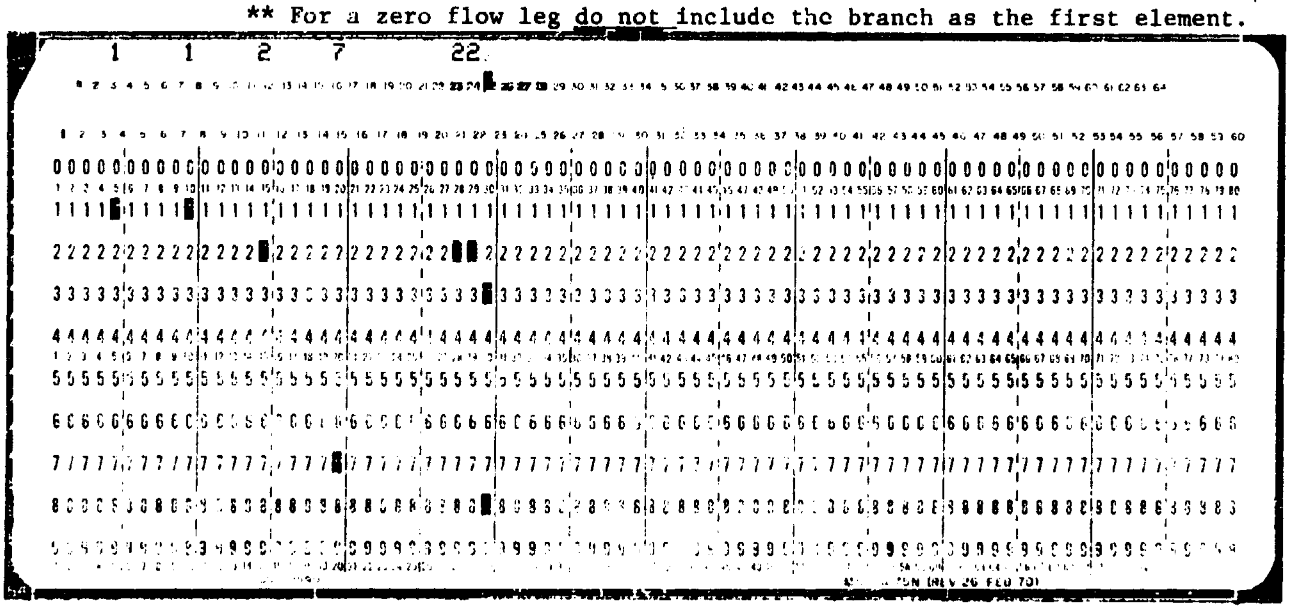
The zero flow leg data cards are repeated for the number of zero flow legs specified on the general data card.

CARD NUMBER 1

COLUMN	FORMAT	DATA	DIMENSIONS
1-5	I5	Leg Number	--
6-10	I5	Upstream Node Number (component # for pressure reference on zero flow leg) *	--
11-15	I5	Downstream Node Number (0 18 zero slow leg)	--
16-20	I5	Number of Elements in Leg	--
21-30	E10.0	Initial Flow Guess	cis
31-40	E10.0	Constant Pressure at Upstream Node	psi
41-50	E10.0	Constant Pressure at Downstream Node	psi
51-60	E10.0		
61-70	E10.0		
71-80	E10.0		

EXAMPLE CARD * The upstream component should be a branch

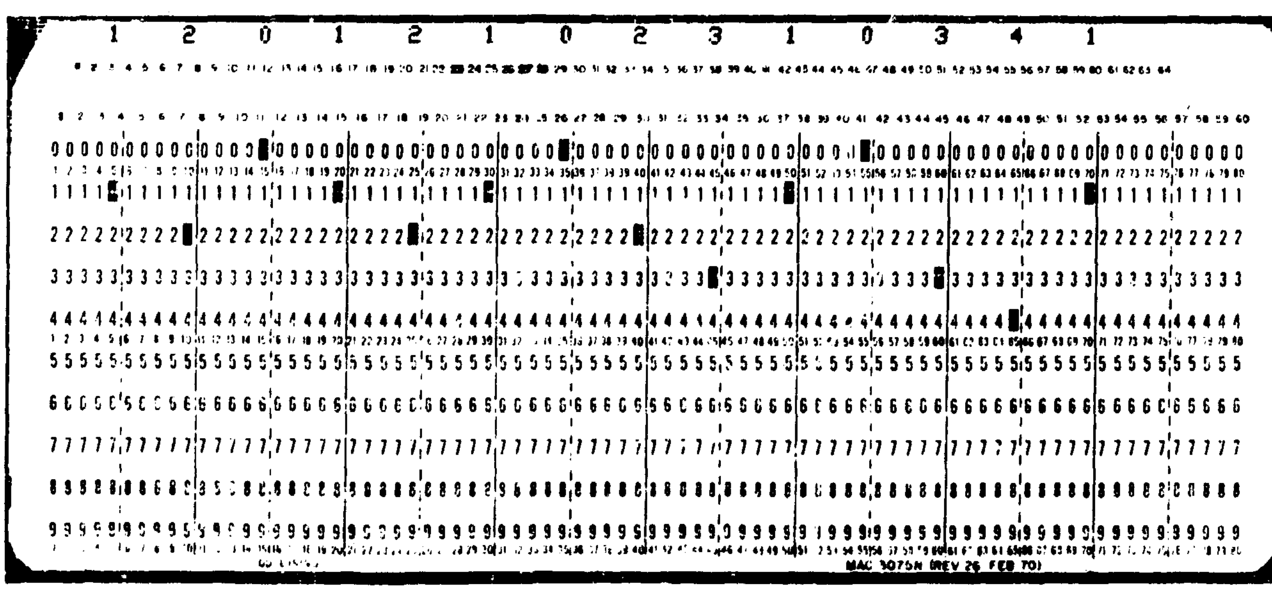
** For a zero flow leg do not include the branch as the first element.



CARD NUMBER 2

COLUMN	FORMAT	DATA
1-5	I5	Component Number or Zero if Line
6-10	I5	Connection or Line Number
11-15	I5	Repeat in Pairs for
16-20	I5	the Number of Elements
21-25	I5	in a Leg - Use as Many
26-30	I5	Cards as Necessary
31-35	I5	
36-40	I5	
41-45	I5	
46-50	I5	
51-55	I5	
56-60	I5	
61-65	I5	
66-70	I5	
71-75	I5	
76-80	I5	

EXAMPLE CARD



APPENDIX L

OUTPUT REQUIREMENTS DATA

HYTRAN USER MANUAL (AFAPL-TR-76-43, VOL. I)

8.0 OUTPUT REQUIREMENTS DATA

The program will output in a print plot form, any calculated system variable versus time. The time interval between plotted points is input on the first general control card.

When using the print plot routine, it should be noted that 101 points are the maximum that can be plotted on one page. When more than 101 points are requested, the plot is continued on an additional page(s).

On the first output data card the user specifies the number of line plot data cards, the number of plotted component variables and specialized outputs which are listed on the plot data card.

The next set of cards are the line variables (paragraph 8.1). The line variables which can be selected are the pressures and flows calculated for each line point.

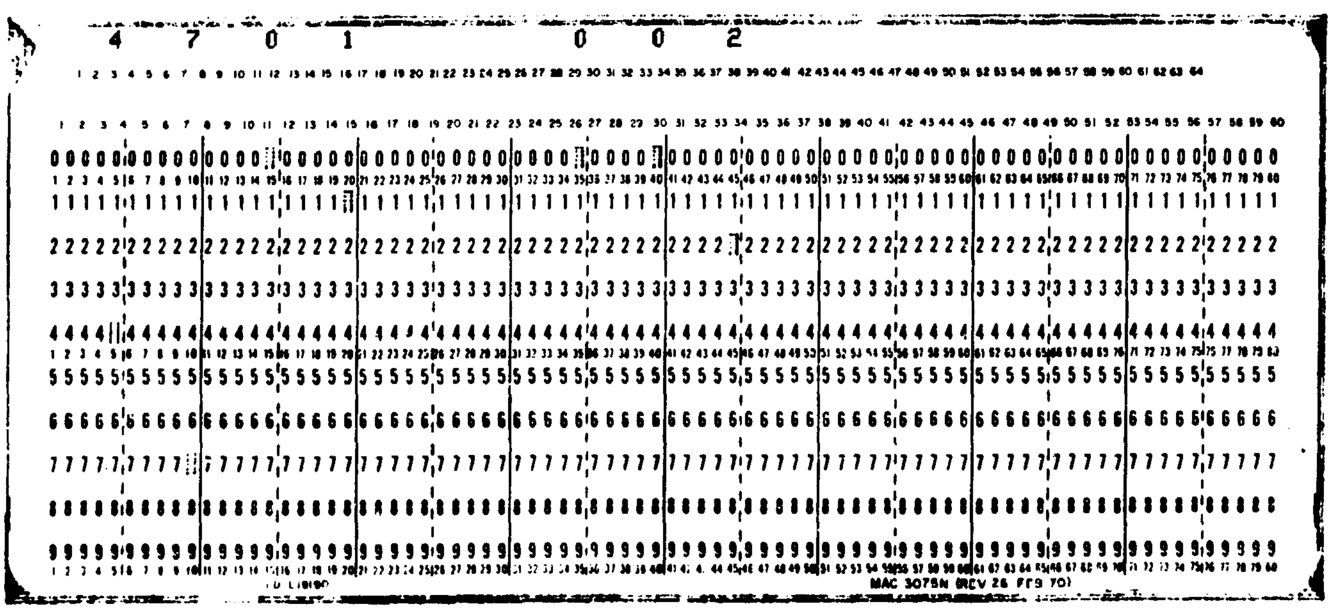
The component variables follow the line cards. The component variables which can be selected are listed in paragraph 8.2.

The Y axis scale can be selected for any print plot. The scale data card is explained in paragraph 8.3. This card follows the component variable cards.

PLOT DATA CARD

COLUMN	FORMAT	DATA
1-5	I5	Number of Line Plot Data Cards
6-10	I5	Number of Component Variables to be Plotted
11-15	I5	+1 - Provides graphs that reflect all maximum values calculated
		-1 - Provides graphs that reflect all minimum values calculated
		0 or - Provides graphs that reflect values calculated at
		Default plot intervals, only
16-20	I5	+1 - Provides a list of all calculated values in addition to
		plots
		0 or - Does not provide a list of calculated values
		Default
21-30		Blank
31-35	I5	+1 - No graphs 0 - Normal graphs
36-40	I5	+1 - Prints absolute value of flow 0 - Normal graphs
41-45	I5	Number of scales

EXAMPLE CARD



8.1 OUTPUT OF LINE VARIABLES

To output pressures and flows at any of the calculated points along a line, the line #, number of plots along the line, and distances along that line from the assumed upstream end have to be input. Unfortunately since the speed of sound varies with temperature, the line is not always divided into the same number of segments.

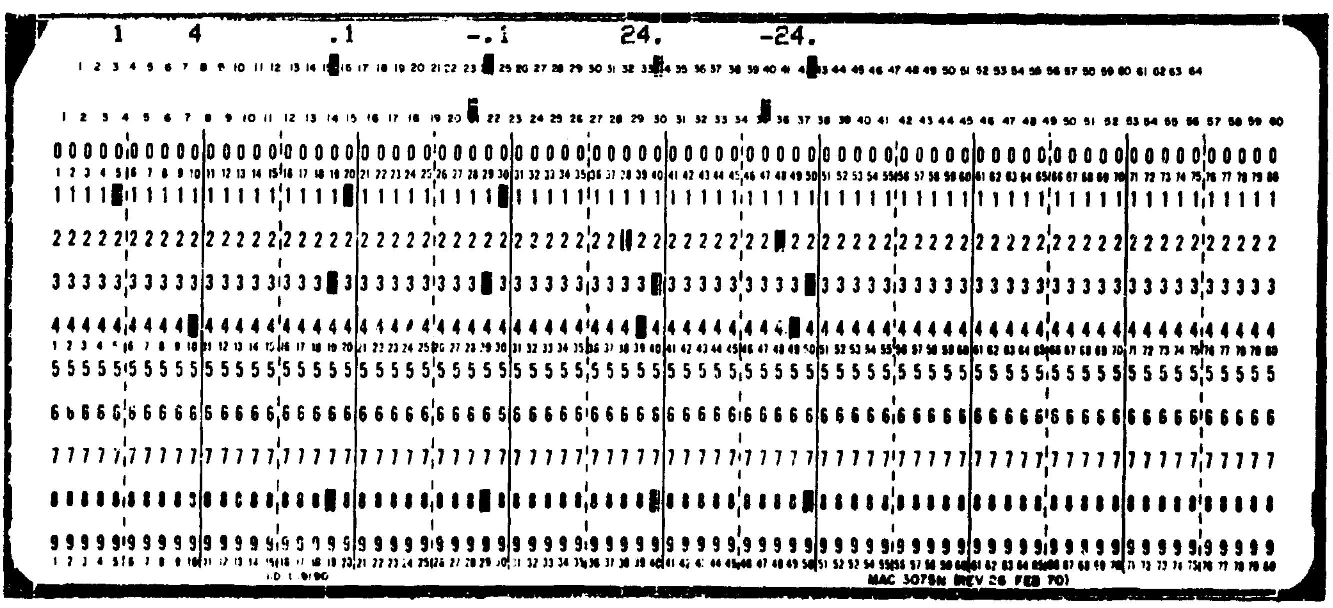
Hence, when a distance along a line is selected, it is unlikely to be a junction point between line segments. The program picks the nearest junction point and outputs on the plot the distance of this junction from the upstream end of the line. The distance is input normally for a pressure plot and as a negative distance for a flow plot. NOTE: The number of cards used must equal the number of "line plot data cards."

LINE PLOT CARD

COLUMN	FORMAT	DATA	DIMENSIONS
1-5	I5	Line Number	
6-10	I5	Number of Plots Along the Line	
11-20	F10.0	Distance Along Line for 1st Plot*	in.
21-30	F10.0	Ditto for up to Seven Points	in.
31-40	F10.0		
41-50	F10.0		
51-60	F10.0		
61-70	F10.0		
71-80	F10.0		

* Distances must be greater than zero.

EXAMPLE CARD



8.2 OUTPUT OF COMPONENT VARIABLES

The component variables to be output are selected from Tables 8.2-1 through 8.2-102.

The total number of component variables to be plotted should equal the number of pairs of data on the following cards.

COMPONENT PLOT CARD

COLUMN	FORMAT	DATA
1-5	I5	Component Number Assigned
6-10	I5	Variable Number to be Plotted
11-15	I5	(This is repeated using additional cards,
16-20	I5	if necessary, until all component variables
21-25	I5	to be plotted have been listed.)
26-30	I5	
31-35	I5	
36-40	I5	
41-45	I5	
46-50	I5	
51-55	I5	
56-60	I5	
61-65	I5	
66-70	I5	
71-75	I5	
76-80	I5	

EXAMPLE CARD

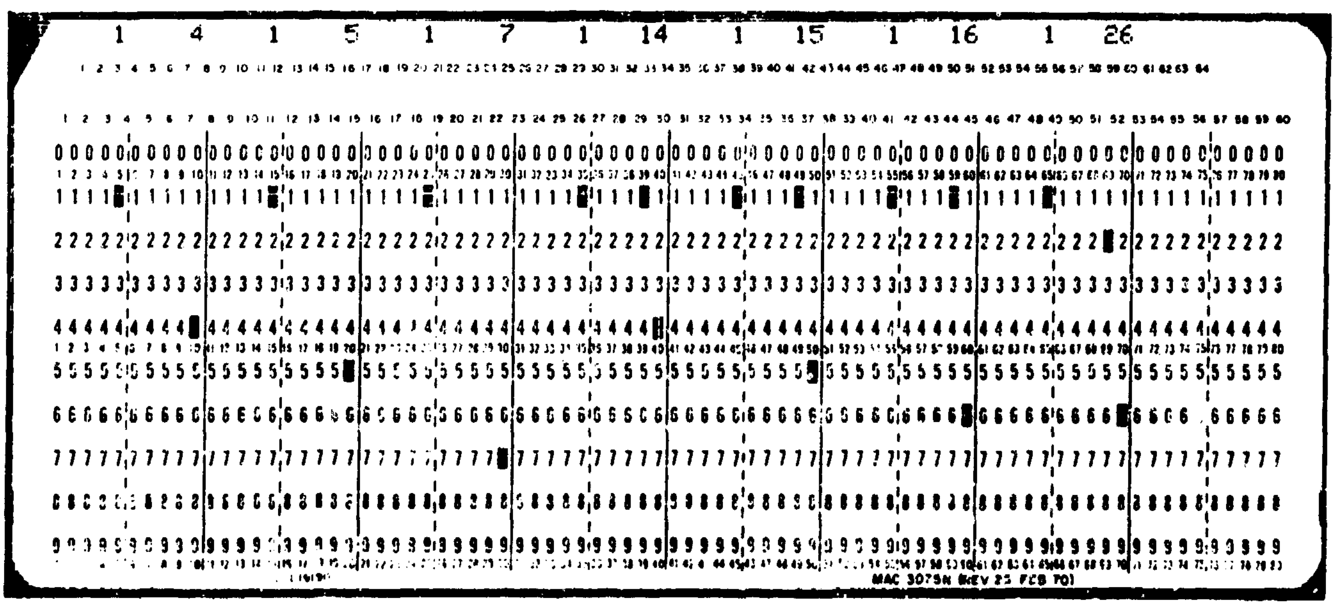


TABLE 8.2-11

BRAN11
PROGRAMMED VARIABLE SELECTION

Number	Name	Description	Dimension
1	--	Cavatation Volume when Multiplied by Calculation Time Interval	in^3

TABLE 8.2-22

VALV22
PROGRAMMED VARIABLE SELECTION

Number	Name	Description	Dimension
8	--	Cavatation Volume - When Multiplied by Calculation Time Interval	in^3

TABLE 8.2-31

CVAL31
PROGRAMMED VARIABLE SELECTION

Number	Name	Description	Dimension
4	VNEW	Poppet Velocity	in/sec
5	ANEW	Poppet Acceleration	in/sec^2
6	XNEW	Poppet Position	in.

TABLE 8.2-51

PUMP51
PROGRAMMED VARIABLE SELECTION

Number	Name	Description	Units
7	PACTU	Pressure in Actuator	psi
14	VELACT	Compensator Actuator Velocity	in/sec
15	DISACT	Compensator Actuator Position	in
16	DISVLV	Compensator Valve Spool Displacement	in
4	QACTU	Flow from Outlet to the Actuator	cis
26	QOUTLT	Net Pumping Flow into Outlet Volume	cis
5	QACTC	Flow from Actuator to the Case	cis
1	PRPM	Pump Speed	rpm
2	PPOWER	Pump Horsepower	hp

TABLE 8.2-54

PUMP54
PROGRAMMED VARIABLE SELECTION

Number	Name	Description	Units
7	PACTU	Pressure in Actuator	psi
14	VELACT	Compensator Actuator Velocity	in/sec
15	DISACT	Compensator Actuator Position	in
16	DISVLV	Compensator Valve Spool Displacement	in
4	QACTU	Flow from Outlet to the Actuator	cis
25	QOUTLT	Net Pumping Flow into Outlet Volume	cis
5	QACTC	Flow from Actuator to the Case	cis
1	PRPM	Pump Speed	rpm
2	TORQUE	Pump Torque	in-lb

TABLE 8.2-62

RSVR62
PROGRAMMED VARIABLE SELECTION

Number	Name	Description	Dimension
9	P2	Reservoir Pressure	psi
1	QNET	Net Reservoir Flow	in^3/sec
10	DUM	Reservoir Volume	in^3

TABLE 8.2-71

ACUM71
PROGRAMMED VARIABLE SELECTION

Number	Name	Description	Dimension
4	PO	Oil Pressure	psi
6	PG	Gas Pressure	psi
8	IVOLO	Oil Vclume	in^3

TABLE 8.2-81

FILT81
PROGRAMMED VARIABLE SELECTION

Number	Name	Description	Dimension
3	PRESSB	Pressure Outside of Element	psi
4	PRESSE	Pressure Inside of Element	psi

TABLE 8.2 - 99

CAD 99

PROGRAMMED VARIABLE SELECTION

Number	Name	Description	Dimension
3	-	Left Outlet Elevon Position	deg
4	-	Left Inbd Elevon Position	deg
5	-	Right Outbd Elevon Position	deg
6	-	Left Inbd Elevon Position	deg
7	-	Rudder Command (rate limited)	deg
8	-	Speedbrake Command (rate limited)	deg
13	-	Angle of attack	deg
14	-	Sideslip angle	deg

TABLE 8.2-101

ACT101
PROGRAMMED VARIABLE SELECTION

Number	Name	Description	Dimension
5	PP1	No. 1 Cylinder Pressure	psi
6	PP2	No. 2 Cylinder Pressure	psi
1	X	Piston Position	in
2	VEL	Piston Velocity	in/sec
18	LOADEX	External Load	lb

TABLE 8.2-102

ACT102
PROGRAMMED VARIABLE SELECTION

Number	Name	Description	Dimension
1	X	Piston Position	in.
2	VEL	Piston Velocity	in./sec
5	LOADEX	External Load	lb.
6	P1	No. 1 Cylinder Pressure	psi
7	P2	No. 2 Cylinder Pressure	psi

Table 8.2 - 106

ACT106
PROGRAMMED VARIABLE SELECTION

Number	Name	Description	Dimension
1	VC	Body Flap Command	--
2	--	Body Flap Position	deg
9	VRT	Body Flap Motor Velocity	rpm

Table 8.2 - 107

ACT107
PROGRAMMED VARIABLE SELECTION

Number	Name	Description	Dimension
1	VC	Rudder command	volts
2	VC+1	Speedbrake command	volts
8	RPOSL	Left panel position	deg
9	RPOSR	Right panel position	deg
12	VRT	Rudder motor velocity	rpm
13	VSBT	Speedbrake motor velocity	rpm

8.3 Selecting Output Plot Scales

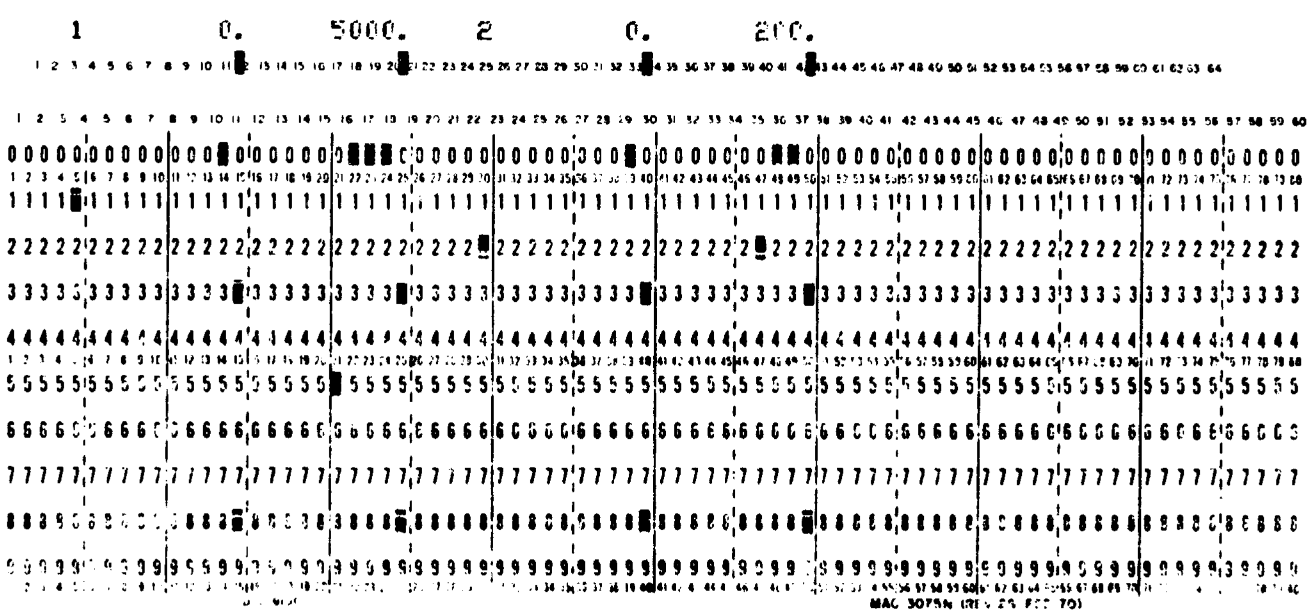
The user has the option of selecting the Y axis scale for any output plot. The scale will remain the same even if the plot is continued on an additional page(s). The graph number must be specified followed by the minimum and maximum Y values as listed on the scale data card. The graph numbering is started in sequence from the first plot entry on the first line data card or the first component variable if there are no line data plots.

Note: When using this option, the maximum Y value cannot be 0.0. If Y max is set to zero the output routine will automatically generate the scale.

SCALE DATA CARD

COLUMN	FORMAT	DATA	DIMENSIONS
1-5	I5	Graph number	
6-15	F10.0	Minimum Y Value	
16-25	F10.0	Maximum Y Value	
		Repeat sequence using additional cards	
		(if necessary) until the number of graphs	
		in columns 41-45 on plot data card is	
		reached.	

EXAMPLE CARD



7.0 OUTPUT SUBROUTINES

The output subroutines comprising, STORE, GRAPH, and SCALED produce print plots of the calculated data.

Current options allow maximum or minimum calculated values to be substituted for plot values in event these max or min values occurred between plot intervals. This assures that the max or min values calculated are reflected in the output plots. Other options list all calculated values for each plot variable, suppress printing of the plots, print absolute values of flow, and allow the user to specify the Y axis scales for any or all printed plots.

7.1 SUBROUTINE STORE

Subroutine STORE, which is called by HYTR, reads output requirements and stores data required for output plots.

7.1.1 Math Model

Not applicable.

7.1.2 Assumptions

Not applicable.

7.1.3 Computation Methods

Section 1000

Section 1000 reads in all the plot information for line and component plots and prints an index of the line and component numbers selected for plotting.

Section 2000

This section first performs a test to determine if the current time step is also a plot time, if so line or component data is stored. If it is not time to store but the MAX/MIN option has been exercised, tests are made to determine if the current calculated value is less than or greater than (depending on which option was exercised) the previous value stored, if so the stored value is replaced by the current calculated value. If the LIST option has been exercised every calculated plot variable is printed. Once all or a max of 101 points have been stored, GRAPH is called to plot the points. A test is then performed to determine if more than 101 points are to be plotted, if so the additional points (up to 101) are calculated and stored as before. GRAPH is again called to plot these points. This procedure is repeated until all points have been plotted.

7.1.4 Approximations

Not applicable.

7.1.5 Limitations

Not applicable.

7.1.6 Variable Name

Variable	Description	Units
I	Counter	
INDEX	Line Number Associated with Pressure and/or Flow Plots	
IPLT	Number of Plots Required along Line INDEX	
IPTS	Dummy Variable	
J	Counter	
LIST	Input Integer Value 0 (No List) of 1 (List of all Points)	
LPT	Distance in DELX's, of Required Plots from Upstream End of Line	
M	Counter	
MXTREM	Dummy Variable	
N	Counter	
NABSQ	Input Integer Value, 0 = Normal Graphs 1 = Prints Absolute Value of Flow for all line Flow Plots	
NAS	Number of Scales Specified by User	
NISTEP	Counter	
NLPLTC	Number of Line Plot Points	
NOGRAF	Input Integer Value 0 = Prints Graphs, 1 = Does Not Print Graphs	
NOMSG	Not Used	
NOSTOP	Not Used	
NPT	Dummy Variable	

7.1.6 (Continued)

Variable	Description	Units
NXTREM	Input Integer Value 0 (Normal Plot), +1 (Plot with Max Values) or -1 (Plot with Min Values	
N1	Counter	--
Y	Dummy Variable	--
YY( )	Array Used to Store Line Positions of Required Plots	--
YSCAL( )	Array to Store User Specified Scales	--

```
      SUBROUTINE STORE
      DOUBLE PRECISION DD
      COMMON NTELPL,NTOLPL,IPT,IPOINT,NPTS,INEL,KNEL,NTOPL,NLPLT(61,3),
     1 VSTORE(1)
      COMMON/SUB/PARM(150,9),PM(1500),QM(1500),P(300),Q(300),C(300)
     1,Z(300),RHO(20),S2ORHO(20),VISC(20),BULK(20),TEMP(20),PVAP(20)
     2,ATPRES,T,DELT,TFINAL,PLTDEL,PI,TITLE(20),LEGN,ICON
     3,KTEMP(99),LSTART(150),NLPT(150),LTYPE(99),NC(99),INX,INZ
     4,INV,ISTEP,NLINE,NEL,IND,IENTR,MNLINE,MNEL,MNLEG,MNNODE,MNPLOT
     5,MNLPTS,MDS,YSCAL(61,2)
      DIMENSION YY(10),DD(1400),ITITLE(40),IN(40),IY(40),IIC(40),IC(40),
     1IITITLE(40),ICHAR(8)
      COMMON/COMPD/D(4500),L(1500),LE(99,4)
      EQUIVALENCE(DD(1),D(1))
      DATA ICHAR/4HLINE,4HDIST,4HCOMP,4HVAR ,2HIN,2H P,2H Q,2H  /
      IF(IENTR) 1000, 1000, 2000
 1000 CONTINUE
      IPT=0
      NISTEP=0
      IPTS=NPTS
      IF(NPTS.GT.101) NPTS=101
C
      NTOLPL=0
      READ(5,270) NLPLTC,NTELPL,NXTREM,LIST,NOSTOP,NOMSG,NOGRAF,NABSQ
     + ,NAS
      IF(NLPLTC.EQ.0) GO TO 142
      DO 140 I=1,NLPLTC
      READ(5,320) INDEX,IPLT,(YY(M),M=1,IPLT)
      DO 130 M=1,IPLT
      J=1
      IF(YY(M).LT.0.0) J=-1
      LPT=(YY(M)*J/PARM(INDEX,5) +0.5)
      IF(LPT.GE.NLPT(INDEX)) LPT =NLPT(INDEX)-1
      NTOLPL=NTOLPL+1
      NLPLT(NTOLPL,2) = INDEX
      NLPLT(NTOLPL,3) = LPT
      NLPLT(NTOLPL,1) = (LSTART(INDEX)+LPT)*J
  130 CONTINUE
  140 CONTINUE
  142 CONTINUE
      NTOPL=NTOLPL+NTELPL
      IF(NTOLPL.GT.MNPLOT) NTOLPL=MNPLOT
C
      IF(NTELPL.EQ.0) GO TO 144
      READ(5,270) ((NLPLT(I+NTOLPL,2),NLPLT(I+NTOLPL,3)),I=1,NTELPL)
  144 CONTINUE
      IF(NTOLPL+NTELPL.GT.MNPLOT) NTELPL=MNPLOT-NTOLPL
      IF(NTOLPL+NTELPL.NE.NTOPL) WRITE(6,520)
      NTOPL=NTOLPL+NTELPL
      IF(NTELPL.EQ.0) GO TO 1250
      LPT=NTOLPL+1
```

```
      DO 1200 I=LPT,NTOPL
      NPT=NLPLT(I,2)
      N  =NLPLT(I,3)
      IF(N) 1150,1180,1160
 1150 N1=-LE(NPT,3)+N+1
      GO TO 1170
 1160 N1=LE(NPT,2)+N-1
 1170 NLPLT(I,1)=N1
      GO TO 1200
 1180 NLPLT(I,1)=1
 1200 CONTINUE
 1250 IF(NAS.GT.0)READ(5,271)((J,YSCAL(J,1),YSCAL(J,2)),I=1,NAS)
  271 FORMAT(3(I5,2F10.0))
      IF(NTELPL.EQ.0)GO TO 1251
C
      WRITE(6,1601)
      WRITE(6,1603)
      IF(NXTREM) 5,12,10
    5 WRITE(6,1606)
      GO TO 1220
   10 WRITE(6,1607)
      GO TO 1220
   12 WRITE(6,1608)
      GO TO 1220
 1220 WRITE(6,1603)
 1251 JJ=0
      II=NTOPL
      DO 1300 I=1,II
      J=I
      JJ=JJ+1
   15 N1=NLPLT(I,1)
      NPT=NLPLT(I,2)
      N=NLPLT(I,3)
      IF(I.GT.NTOLPL) GO TO 2500
      ITITLE(JJ)=ICHAR(1)
      IN(JJ)=NPT
      IY(JJ)=N*PARM(NPT,5)
      IITITLE(JJ)=ICHAR(2)
      NI=0
      IF(N1.LT.0) NI=1
      IC(JJ)=ICHAR(6+NI)
      IIC(JJ)=ICHAR(5)
      GO TO 1700
 2500 ITITLE(JJ)=ICHAR(3)
      IITITLE(JJ)=ICHAR(4)
      IN(JJ)=NPT
      IY(JJ)=N
      IC(JJ)=ICHAR(8)
      IIC(JJ)=ICHAR(8)
 1700 IF(JJ.LT.10.AND.I.LT.NTOPL) GO TO 1300
      III=I-JJ+1
```

```
      WRITE(6,1500) ((JJJ),JJJ=III,J)
      IF(I.GT.NTOLPL) GO TO 75
      WRITE(6,1600) ((ITITLE(JJJ),IN(JJJ),IC(JJJ)),JJJ=1,JJ)
      WRITE(6,1600) ((IITITLE(JJJ),IY(JJJ),IIC(JJJ)),JJJ=1,JJ)
      GO TO 16
   75 WRITE(6,1604) ((ITITLE(JJJ),IN(JJJ),IC(JJJ)),JJJ=1,JJ)
      WRITE(6,1604) ((IITITLE(JJJ),IY(JJJ),IIC(JJJ)),JJJ=1,JJ)
   16 WRITE(6,1605)
      JJ=0
 1300 CONTINUE
      WRITE(6,1602)
      WRITE(6,1603)
 2000 CONTINUE
C
      IF(ISTEP.EQ.NISTEP) GO TO 2010
      IF(NXTREM.EQ.0.AND.LIST.EQ.0) RETURN
      MXTREM=NXTREM
      GO TO 2020
 2005 NISTEP=NISTEP-IPOINT
 2010 MXTREM=0
      IPT=IPT+1
      NISTEP=NISTEP+IPOINT
      VSTORE(IPT)=T
 2020 NPT=IPT
      N1=0
      DO 2200 I=1,NTOPL
      NPT=NPT+NPTS
      N=NLPLT(I,1)
      IF(I.GT.NTOLPL) GO TO 2050
      IF(N) 2030,2150,2040
 2030 Y=QM(-N)
      GO TO 2080
 2040 Y=PM(N)
      GO TO 2080
 2050 IF(N) 2060,2150,2070
 2060 Y=DD(-N)
      GO TO 2080
 2070 Y=D(N)
 2080 IF(MXTREM)2090,2085,2100
 2085 IF(ISTEP+IPOINT.EQ.NISTEP) GO TO 2110
      GO TO 2120
 2090 IF(VSTORE(NPT).GT.Y) GO TO 2110
      GO TO 2120
 2100 IF(VSTORE(NPT).GE.Y) GO TO 2120
 2110 VSTORE(NPT)=Y
 2120 IF(LIST.EQ.0) GO TO 2200
      IF(I.EQ.1) WRITE(6,2211) T
      N1=N1+1
      YY(N1)=Y
      IF(N1.EQ.10.OR.I.EQ.NTOPL)GO TO 2130
      GO TO 2200
```

```
 2130 WRITE(6,2210)(YY(JJ),JJ=1,N1)
      N1=0
      GO TO 2200
 2150 WRITE(6,2220) I
      Y=T
      GO TO 2080
 2200 CONTINUE
C     IF(N1*LIST.NE.0)WRITE(6,2210) (YY(I),I=1,N1)
      IF(IPT.NE.NPTS) RETURN
      INV=NABSQ
      IF(NOGRAF.EQ.0)CALL GRAPH
      IF(IPTS-NPTS) 2300,2350,2310
 2300 NPTS=IPTS
      IPT=0
      GO TO 2005
 2310 NPTS=101
      IPTS=IPTS-100
      IPT=0
      GO TO 2005
 2350 CONTINUE
      RETURN
  270 FORMAT (16I5)
  320 FORMAT (2I5,7F10.0)
  520 FORMAT(5X,42HTOO MANY PLOTS REQUESTED MAX NUMBER IS 60     )
 2210 FORMAT(5X,10E12.5)
 2211 FORMAT(5X,25HDATA CALCULATED AT TIME =,F8.4)
 2220 FORMAT(5X,45HVALUE OF N IN 2000 SECTION OF COMP IS ZERO I=  ,I5)
 1500 FORMAT(5X,10(6HGRAPH ,I4,2H  ))
 1600 FORMAT(5X,10(A4,I4,A2,2H  ))
 1601 FORMAT(1H1,42X,35HVARIABLES SELECTED FOR OUTPUT PLOTS)
 1602 FORMAT(1H1,53X,13HHYTRAN OUTPUT)
 1603 FORMAT(1H0)
 1604 FORMAT(5X,10(A4,I6,A2))
 1605 FORMAT(1H )
 1606 FORMAT(28X,71HVALUES PLOTTED REPRESENT MINIMUM VALUES CALCULATED I
     1N THE TIME INTERVAL)
 1607 FORMAT(28X,71HVALUES PLOTTED REPRESENT MAXIMUM VALUES CALCULATED I
     2N THE TIME INTERVAL)
 1608 FORMAT(21X,75HVALUES PLOTTED REPRESENT THE ACTUAL VALUES CALCULATE
     3D AT EACH PLOT INTERVAL)
      END
```

7.2 SUBROUTINE GRAPH

Subroutine graph produces print plots of the output data stored in VSTORE ( ).

Most computers will have their own version of this subroutine which could be used if necessary. However, since the plotted output is such an integral part of HYTRAN, this subroutine has been added to avoid the problems involved in changing from one computer to another.

7.2.1 Theory - Not applicable.

7.2.2 Assumptions - Not applicable.

7.2.3 Limitations

The program is executed once for each plot, up to the total number of plots NTOPL. The DO 901 J = 1, NTOPL controls this loop.

The first section which sets the X scale, is only executed on the first pass, when J = 1.

The program currently uses TIME (1) as XMIN and TIME (NPTS) as XMAS.

In the second section a DO loop is used to find the maximum and minimum values of the Y data to be plotted, using the functions AMAX1 (YMAY, VSTORE, (I+IADD)) and AMIN1 (YMIN, VSTORE (I+IADD)).

With the maximum and minimum values established, a check is made to see if they are equal, if they are, 25 is added to YMAX, and YMIN is set at 50 less than that, to avoid a fruitless search for a suitable scale.

Subroutine SCALED is then called to obtain a preferred scale for the Y axis, and returns with values for YMAX and YMIN.

A check is made to determine if the user specified YMAX and YMIN.

IF (YSCAL(J,2).NE.O.) GO TO 2000

If the above test passes, the YMIN and YMAX values are set to the input values and the subroutine execution continues.

The next section finds the type of plot and sets the plot character, P, Q or C and the data to be written at the bottom of the output plot.

The routine starts the output plot section by going to the top of a new page, and proceeds to plot the output data, line by line until the plot is complete.

At the bottom of the plot a descriptive line is written which gives the line number and distance along the line for line pressure or flow plots or the variable number and the component number if it is a component data plot.

The next printed line is the title of the run input on the first data card.

When all the plots have been completed program control returns to HYTR.

7.2.4 Approximations - Not applicable.

7.2.5 Limitations

The basic limitation of a print plot is the number of points that can be plotted on a single page graph and the resulting inaccuracy in reading the graph. To an extent these limitations can beovercome by use of the MAX/MIN and LIST options noted in Section 2.4 of Volume I of this report.

7.2.6 Variable Names

Variable	Description	Units
AVS	Absolute value of VS	
DIST	Distance of Plot Point Down a Line	IN
I	Counter	
IADD	Address J*NPTS	
ICHAR	Plot character	
ICHAR( )	X and Y Axis Write Characters	
ISP	Counter	
ISPACE( )	Temporary Variable for Writing X and Y Axis Scales	
ITEST	Counter	
J	Counter Indicating Plot Number	
L	Dummy Variable	
LINE	Integer Counter for Plot Line Number	
NABSQ	Integer value 1 or 0 Used as Indicator	
NCHAR	Dummy Variable Representing Plot Character	
NVAR	Dummy Variable Representing Point at which Line Plot is taken or Component Number	
SP	Column Number Nearest to the Ith Value of X-Variable	
VS	Dummy Variable	

Variable	Description	Units
XAX	Temporary Variable for Writing X Axis Scale Values	
XDELTA	Distance Between Stored Points on X Axis	
XMAX	Last (Largest) X Axis Value	
XMIN	First (Lowest) X Axis Value	
XSCALE	X Scale Range	
Y	Temporary Variable (Y Axis Scale Value)	
YDELTA	Distance Between Stored Points on the Y Axis	
YLAST	Last Y Axis Scale Value	
YLO	Lowest Value in Search Range	
YMAX	Maximum Value to be Plotted	
YMIN	Minimum Value to be Plotted	
YUP	Highest Value in Search Range	
YSCAL( )	Array Contain Input YMIN and YMAX Values	

```
      SUBROUTINE GRAPH
C *** REVISED APRIL 27,1979
C *** THIS SUBROUTINE IS A MODIFICATION OF THE STANDARD HYTRAN GRAPH
C *** SUBROUTINE. WITH THIS SUBROUTINE, IF MORE THAN ONE SET OF GRAPHS
C *** IS PLOTTED, THE Y - SCALE REMAINS THE SAME ON ALL SETS
      COMMON NTELPL,NTOLPL,IPT,IPOINT,NPTS,INEL,KNEL,NTOPL,NLPLT(61,3),
     1 VSTORE(1)
      COMMON/SUB/PARM(150,9),PM(1500),QM(1500),P(300),Q(300),C(300)
     1,Z(300),RHO(20),S2ORHO(20),VISC(20),BULK(20),TEMP(20),PVAP(20)
     2,ATPRES,T,DELT,TFINAL,PLTDEL,PI,TITLE(20),LEGN,ICON
     3,KTEMP(99),LSTART(150),NLPT(150),LTYPE(99),NC(99),INX,INZ
     4,INV,ISTEP,NLINE,NEL,IND,IENTR,MNLINE,MNEL,MNLEG,MNNODE,MNPLOT
     5,MNLPTS,MDS,YSCAL(61,2)
      DIMENSION ISPACE(101),XAX(6),ICHART(8)
      DATA ICHART/1HQ,1HP,1HC,1HI,1H-,1H+,1H ,1H*/
      DATA ITEST,XSCALE/0,0.0/
C-----BEGIN OUTER LOOP. FIND X PARAMETERS ON FIRST PASS ONLY
    1 DO 901 J=1, NTOPL
      IADD=J*NPTS
      IF(J.NE.1) GO TO 2
      XMAX=VSTORE(NPTS)
      XMIN=VSTORE(1)
      ITEST=ITEST+1
      IF(ITEST.NE.1) XMAX=XMIN+XSCALE
      XSCALE=XMAX-XMIN
C     CALL SCALED(XMAX,XMIN)
      XDELTA=(XMAX-XMIN)/100.
C-----FIND Y PARAMETERS
C-----TEST IF SECOND SET OF GRAPHS
    2 CONTINUE
      IF(YSCAL(J,2).NE.0.0)GO TO 2000
      YMAX=VSTORE(1+IADD)
      YMIN=YMAX
      DO 902 I=2, NPTS
      YMAX=AMAX1(YMAX,VSTORE(I+IADD))
  902 YMIN=AMIN1(YMIN,VSTORE(I+IADD))
      NABSQ=0
      IF(J.GT.NTOLPL.OR.NLPLT(J,1).GE.0) GO TO 905
      IF(INV.EQ.0) GO TO 905
      IF(YMIN.GT.0) GO TO 905
      NABSQ=1
      IF(ABS(YMIN).GT.YMAX) YMAX=ABS(YMIN)
  905 IF(YMAX.NE.YMIN)GO TO 9020
      YMAX=YMAX + 25.
      YMIN=YMAX - 50.
      GO TO 9025
 9020 AMAX = (YMAX+YMIN)*.001
      IF((YMAX-YMIN).GT.AMAX) GO TO 9025
      YMAX = YMAX+AMAX
      YMIN = YMIN-AMAX
 9025 CALL SCALED(YMAX,YMIN)
      GO TO 2010
```

```
 2000 YMAX=YSCAL(J,2)
      YMIN=YSCAL(J,1)
 2010 CONTINUE
      YDELTA=(YMAX-YMIN)/50.
C-----FIND LINE/COMPONENT NUMBER, TYPE OF PLOT, OUTPUT DATA
      L=NLPLT(J,2)
      IF (J.GT.NTOLPL) GO TO 5
      IF(NLPLT(J,1)) 3,5,4
    3 ICHAR=ICHART(1)
      DIST=NLPLT(J,3)*PARM(L,5)
      GO TO 6
    4 ICHAR=ICHART(2)
      DIST=NLPLT(J,3)*PARM(L,5)
      GO TO 6
    5 ICHAR=ICHART(3)
      NVAR=NLPLT(J,3)
C-----GO TO TOP OF NEXT PAGE
    6 WRITE(6,601)
C-----LOOP FOR EACH PLOT LINE.
      Y=YMAX + YDELTA
    7 DO 907 LINE=1, 51
      YLAST=Y
      Y=Y-YDELTA
      YUP=Y+YDELTA/2.
      YLO=Y-YDELTA/2.
C-----FIRST + LAST CHAR. ON LINE = *I*
      ISPACE(1)=ICHART(4)
      ISPACE(101)=ICHART(4)
C-----FIRST + LAST LINES ALL *-*, EXCEPT *+* IN 11,21,31,41,...,81,+91
      IF(LINE.NE.1 .AND. LINE.NE.51)GO TO 11
    9 DO 909 ISP=2,100
      IF((ISP-1).EQ.(ISP-1)/10*10)GO TO 10
      ISPACE(ISP)=ICHART(5)
      GO TO 909
   10 ISPACE(ISP)=ICHART(6)
  909 CONTINUE
      GO TO 14
C-----INITIAIZE COL. 2-100 ON LINES 2-50 TO * *, OR *+----+-* IF AXIS
   11 IF(Y.LE.0. .AND.YLAST.GT.C.)GO TO 13
   12 DO 912 ISP=2, 100
  912 ISPACE(ISP)=ICHART(7)
      GO TO 14
   13 DO 913 ISP=2,100
      ISPACE(ISP)=ICHART(5)
  913 IF((ISP-1).EQ.(ISP-1)/10*10)ISPACE(ISP)=ICHART(6)
C---- SEARCH Y-VALUE ARRAY FOR THOSE IN RANGE YLO.LT.VALUE.GE.YUP
   14 DO 914 I=1, NPTS
      VS=VSTORE(I+IADD)
      NCHAR=ICHAR
      IF(VS.GT.YLO .AND. VS.LE.YUP)GO TO 145
      IF(NABSQ.NE.1) GO TO 914
```

```
      AVS=ABS(VS)
      IF(AVS.LT.YLO.OR.AVS.GT.YUP) GO TO 914
      NCHAR=ICHART(8)
C-----FIND COLUMN NUMBER NEAREST TO I-TH VALUE OF X-VARIABLE WHEN SCALED
  145 SP=(VSTORE(I)-XMIN)/XDELTA + 1
      IF(SP-AINT(SP).GT.0.50) SP=SP + 0.50
      ISP=SP
C-----CHECK ISP. IF LT 0 OR GT 102, ERROR; IF 0, ADD 1; IF 102, SUBT. 1
      IF(ISP) 914, 15, 16
   15 ISP=1
      GO TO 18
   16 IF(ISP-102)18,17,914
   17 ISP=101
   18 ISPACE(ISP)=NCHAR
  914 CONTINUE
C-----LINES 1,11,21,31,41,+51 HAVE Y-VALUES; THESE LINES, PLUS LINES 6,
C     16,26,... ALSO HAVE *+* IN COL. 1+101 IF EMPTY
      IF((LINE-1).NE.(LINE-1)/5*5)GO TO 19
      IF(ISPACE(1).NE.ICHAR) ISPACE(1)=ICHART(6)
      IF(ISPACE(101).NE.ICHAR)ISPACE(101)=ICHART(6)
      IF((LINE-1).NE.(LINE-1)/10*10)GO TO 19
C-----WRITE OUT PLOT LINE, CONTINUE
      WRITE(6,602)Y, ISPACE
      GO TO 907
   19 WRITE(6,603)ISPACE
  907 CONTINUE
C-----CALCULATE + PRINT X-AXIS VALUES
   20 DO 920 I=1, 6
  920 XAX(I)=XMIN + (I-1)*20.*XDELTA
      WRITE(6,604) XAX
C-----WRITE LOWER TITLES + VALUES, REENTER OUTER LOOP
      IF (J.GT.NTOLPL) GO TO 23
      IF(NLPLT(J,1))21,23,22
   21 WRITE(6,605) J,DIST,L
      GO TO 900
   22 WRITE(6,606) J,DIST,L
      GO TO 900
   23 WRITE(6,607) J,NVAR,L
  900 CONTINUE
      WRITE(6,608)TITLE
  901 CONTINUE
 1000 WRITE(6,601)
      WRITE(6,610)
      WRITE(6,611)
      IF(T.LT.TFINAL-DELT)RETURN
      DO 1250 J=1,NTOPL
      L=NLPLT(J,2)
      IF(J.GT.NTOLPL) GO TO 50
      DIST=NLPLT(J,3)*PARM(L,5)
```

```
      IF(NLPLT(J,1)) 25,1250,30
   25 WRITE(6,605) J,DIST,L
      GO TO 1250
   30 WRITE(6,606) J,DIST,L
      GO TO 1250
   50 NVAR=NLPLT(J,3)
      WRITE(6,607) J,NVAR,L
 1250 CONTINUE
  601 FORMAT(1H1)
  602 FORMAT(1X,15X,F12.4,1X,101A1)
  603 FORMAT(1X,28X,101A1)
  604 FORMAT(1X,23X,5(F9.3,11X),F9.3)
  605 FORMAT(1X,28X,6HGRAPH ,I3,1X,53H  FLOW (CU.IN/SEC) VS. TIME (SEC.)
     + FOR A DISTANCE OF ,F8.2,26H INCHES ALONG LINE NUMBER ,I5)
  606 FORMAT(1X,28X,6HGRAPH ,I3,1X,53H  PRESSURE  (PSIA) VS. TIME (SEC).
     +FOR A DISTANCE OF ,F8.2,26H INCHES ALONG LINE NUMBER ,I5)
  607 FORMAT(1X,28X,6HGRAPH ,I3,1X,18H  VARIABLE NUMBER ,I3,21H OF COMPO
     +NENT NUMBER ,I3,38H VS. TIME (SEC.).  THE VARIABLE IS ---  )
  608 FORMAT(1X,28X,20A4)
  610 FORMAT(1H0,65X,27HHYTRAN PROGRAM OUTPUT PLOTS)
  611 FORMAT(1H0)
      RETURN
      END
```

```
      SUBROUTINE SCALED(RMAX,RMIN)
      DIMENSION SCALE(6)
      DATA SCALE/.5,1.,2.,5.,10.,20./
C-----FIND THE RANGE OF VALUES *RANGE*, AND PLACE ACTUAL MAX AND MIN
C-----POINTS IN *AMAX* AND *AMIN*
      RANGE=RMAX-RMIN
      AMAX=RMAX
      AMIN=RMIN
C-----FIND AN INTEGER EXPONENT *IEXP* AND BASE *MANT* SUCH THAT THE
C-----VALUE OF MANT**IEXP IS .GE. RANGE
      IEXP=ALOG10(RANGE)
      MANT=RANGE/10.**IEXP
      IF(RANGE.GT.MANT*10.**IEXP)MANT=MANT+1
C-----USING MANT, SELECT ONE OF THE PREFERRED SCALES
      IF(MANT.GT.10)GO TO 70
      IF(MANT.LT.1) GO TO 70
      GO TO(80,90,100,100,100,110,110,110,110,70), MANT
   70 MANT=1
      IEXP=IEXP+1
   80 J=2
      GO TO 120
   90 J=3
      GO TO 120
  100 J=4
      GO TO 120
  110 J=5
C-----SET *IEMAX* EQUAL TO THE EXPONENT OF 10. CORRESPONDING TO RMAX
  120 IF(RMAX.EQ.0.) GO TO 121
      IEMAX=ALOG10(ABS(RMAX))
C-----USE AMAX AND IEMAX TO FIND A POSSIBLE MAXIMUM VALUE FOR THE
C-----SCALE.  PLACE THE VALUE IN RMAX, AND COMPARE WITH THE ACTUAL
C-----MAXIMUM POINT.
      RMAX=INT(ABS(AMAX)/10.**IEMAX)*10.**IEMAX*SIGN(1.0,AMAX)
  121 IF(RMAX.GE.AMAX)GO TO 130
C-----IF RMAX IS .LT. ACTUAL MAX POINT, INCREASE IT BY 5
C-----PERCENT AND RECHECK--REPEAT AS NECESSARY
      RMAX=RMAX+.05*SCALE(J)*10.**IEXP
      GO TO 121
C-----SET THE SCALE'S MINIMUM BY SUBTRACTING SCALE(J)**IEXP FROM RMAX
C-----IF THE ACTUAL MINIMUM *AMIN* LIES WITHIN THE RANGE NOW DEFINED
C-----BY RMAX AND RMIN, CONTINUE.
  130 RMIN=RMAX-SCALE(J)*10.**IEXP
      IF(RMIN.LE.AMIN)GO TO 150
C-----GO TO THE NEXT LARGEST SCALE, RECALCULATE RMIN, AND RECHECK
      J=J+1
      IF(J.LT.5.5)GO TO 130
      J=1
      IEXP=IEXP+1
      GO TO 130
```

```
C-----IF THE SCALE'S MIN IS .LT. ZERO, BUT THE ACTUAL MIN IS POSITIVE,
C-----SHIFT THE SCALE UP SO THAT THE SCALE BEGINS AT ZERO
  150 IF(RMIN*AMIN.GT.0.)GO TO 170
      RMIN=0.
  160 RMAX=SCALE(J)*10.**IEXP
C-----DUE TO THE SHIFT, IT MAY BE POSSIBLE TO DECREASE THE SCALE TO
C-----THE NEXT SMALLEST SIZE
      IF(AMAX.GT.1.000001*SCALE(J-1)*10.**IEXP) RETURN
      J=J-1
      IF(J.GT.1.5) GO TO 160
      J=4
      IEXP=IEXP-1
      GO TO 160
C-----IF RMIN IS POSITIVE AND NEAR ZERO, SHIFT THE SCALE DOWN TO 0. MIN
  170 IF(RMIN.LT.0.)GO TO 175
      IF(RMIN.GT..1*RMAX)GO TO 180
      RMIN=0.
      RMAX=SCALE(J)*10.**IEXP
C-----IF THE SHIFT DOWN CAUSES RMAX TO LIE BELOW THE ACTUAL MAX,
C-----INCREASE THE SCALE RANGE TO THQ NEXT LARGEST
      IF(RMAX.LT.AMAX)RMAX=SCALE(J+1)*10.**IEXP
      RETURN
C-----IF RMAX IS NEGATIVE AND NEAR ZERO, SHIFT THE SCALE UP TO 0. MAX
 175  IF(RMAX.GE.0.) GO TO 180
      IF(-RMAX.GT.-0.1*RMIN) GO TO 180
      RMAX=0.
      RMIN=-SCALE(J)*10.**IEXP
C-----IF THE SHIFT UP CAUSES RMIN TO LIE ABOVE THE ACTUAL MIN,
C-----INCREASE THE SCALE RANGE TO THE NEXT LARGEST
      IF(RMIN.GT.AMIN) RMIN=-SCALE(J+1)*10.**IEXP
      RETURN
C-----CENTER THE SCALE ABOUT THE ACTUAL RANGE OF POINTS
  180 ITOP=(RMAX-AMAX)/(.05*SCALE(J)*10.**IEXP)
      IBOT=(AMIN-RMIN)/(.05*SCALE(J)*10.**IEXP)
      IDIF=(ITOP-IBOT)/2
      IF(IDIF.EQ.0) RETURN
      RMIN=RMIN-IDIF*.05*SCALE(J)*10.**IEXP
      RMAX=RMIN+SCALE(J)*10.**IEXP
      RETURN
      END
```

APPENDIX N
FUNCTION SECORD
HYTRAN TECHNICAL MANUAL (AFAPL-TR-76-43, VOL II)

8.11 FUNCTION SECORD

Function SECORD computes the time response of a second order function to either a unit step or impulse forcing function.

8.11.1 Math Model

A block diagram for a second order system is shown in Figure 8.11-1.

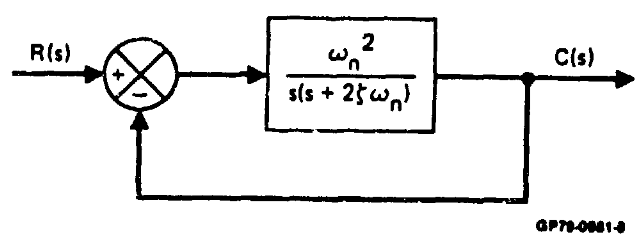


FIGURE 8.11-1 SECOND ORDER BLOCK DIAGRAM

The closed loop transfer function $\frac{C(s)}{R(s)}$ can be written

$$\frac{C(s)}{R(s)} = \frac{\omega_n^2}{s^2 + 2\zeta\omega_n s + \omega_n^2} \qquad (1)$$

The dynamic behavior of a second order system can be described in terms of two parameters, ω_n - the system natural frequency and, ζ- the damping factor.

Unit Step

If R(s) is a unit step, Equation (1) can be written as

$$C(s) = \frac{\omega_n^2}{s^2 + 2\zeta\omega_n s + \omega_n^2} * \frac{1}{s} \qquad (2)$$

Equation (2) can be expanded to

$$C(s) = \frac{1}{s} - \frac{s + \zeta\omega_n}{s^2 + 2\zeta\omega_n s + \omega_n^2} - \frac{\zeta\omega_n}{s^2 + 2\zeta\omega_n s + \zeta_n^2} \qquad (3)$$

let $\omega_d^2 = \omega_n^2 (1 - \zeta^2)$

Then Equation (3) becomes

$$C(s) = \frac{1}{s} - \frac{s + \zeta\omega_n}{(s + \zeta\omega_n)^2 + \omega_d^2} - \frac{\zeta\omega_n}{(s + \zeta\omega_n)^2 + \omega_d^2} \quad (4)$$

Taking the Laplace transform of Equation (4) assuming zero initial conditions yields

$$c(t) = 1 - \frac{e^{-\zeta\omega_n t}}{\sqrt{1 - \zeta^2}} \sin\left(\omega_d t + \tan^{-1} \frac{\sqrt{1-\zeta^2}}{\zeta}\right) \quad (5)$$

Where $t \geq 0$ and $0 < \zeta < 1$

When ζ is equal to zero, the response is undamped and the oscillations continue indefinitely. A damping coefficient greater than or equal to one results in an undefined response. Equation (5) can be approximated and solved for the critically ($\zeta = 1$) and overdamped cases ($\zeta > 1$). The response is similar to that of a first order system.

<u>Impulse</u>

For a unit-inpulse input R(s) is equal to 1 or

$$C(s) = \frac{\omega_n^2}{s^2 + 2\zeta\omega_n s + \omega_n^2} \quad (6)$$

The inverse Laplace transform of Equation (6) gives the time response for $0 \leq \zeta < 1$ as

$$c(t) = \frac{\omega_n}{\sqrt{1 - \zeta^2}} e^{-\zeta\omega_n t} \sin \omega_n \sqrt{1-\zeta^2} t \quad (7)$$

Where $t \geq 0$

8.11.2 Computation Method

The impulse and unit step responses of a second order function are initialized to zero by setting the X argument to zero. When the response is desired, X must be set to a number in the calling program. Time is incremented from zero by the selected delta interval (DT) on subsequent calls to the SECORD function.

Equation (5) is used to compute the time response of a second order function to a unit step. Equation (7) calculates the response to an impulse input.

8.11.3 Variable Listing

NAME	DESCRIPTION	DIMENSIONS
A,B,C	Dummy Variables	-
ARR	Current Time	Sec
DT	Time Step	Sec
J	Input Forcing Function J = 1 Unit Step J = 2 Impulse	-
OMEGA	Undamped Natural Frequency	RAD/SEC
WD	Damped Natural Frequency	RAD/SEC
X	Response Indicator X = 0, No Response X NE 0, Response	-
ZETA	Damping Factor	-

8.11.4 Subroutine Listing

```
      FUNCTION SECORD(J,X,OMEGA,ZETA,DT,ARR)
      IF(X.EQ.0.)GO TO 5
      ARR=ARR+DT
      IF(J.EQ.1)GO TO 10
      IF(J.EQ.2)GO TO 20
 5    SECORD=0.
      ARR=0.
      RETURN
10    CONTINUE
      A=OMEGA*ZETA*ARR
      B=SQRT(1.-ZETA*ZETA)
      WD=OMEGA*B
      C=EXP(-A)/B*SIN(WD*ARR+ATAN(B/ZETA))
      SECORD=1.-C
      RETURN
20    CONTINUE
      A=SQRT(1.-ZETA*ZETA)
      B=SIN(OMEGA*A*ARR)
      C=EXP(-ZETA*OMEGA*ARR)
      SECORD=(1./A)*B*C
      RETURN
      END
```

APPENDIX O
BLOCK DATA AND COMP SUBROUTINES
HYTRAN TECHNICAL MANUAL (AFAPL-TR-76-43, VOL. II)

BLOCK DATA SUBROUTINE

```
      BLOCK DATA
C *** REVISED MARCH, 1979 ***
      COMMON/SUB/PARM(150,9),PM(1500),QM(1500),P(300),Q(300),C(300)
     1,Z(300),RHO(20),S2ORHO(20),VISC(20),BULK(20),TEMP(20),PVAP(20)
     2,ATPRES,T,DELT,TFINAL,PLTDEL,PI,TITLE(20),LEGN,ICON
     3,KTEMP(99),LSTART(150),NLPT(150),LTYPE(99),NC(99),INX,INZ
     4,INV,ISTEP,NLINE,NEL,IND,IENTR,MNLINE,MNEL,MNLEG,MNNODE,MNPLOT
     5,MNLPTS,MDS,YSCAL(61,2)
      COMMON /COMPD/ D(4500),LT(100),L11(10),L1220(90),L21(10),L22(10)
     + L23(10),L24(10),L25(10),L2630(50),
     + L31(10),L32(10),L33(10),L34(10),L3540(60),L41(10),L42(10),
     + L4350(80),L51(10),L52(10),L53(10),L54(10),
     + L55(10),L56(10),L57(10),L5860(30),
     + L61(10),L62(10),L63(10),L6470(70),L71(10),L72(10),L73(10),
     + L7480(70),L81(10),L82(10),L83(10),L8490(70),L91(10),L9293(20),
     + L9495(20),L9697(20),L98(10),L99100(20),
     + L101(10),L102(10),L103(10),L104(10),L105(10),L106(10),L107(10),
     + L108(10),L109(10),L110(10),LEND(400),LE(99,4)
      DATA MNLINE,MNEL,MNLEG,MNNODE,MNPLOT,MNLPTS,MDS,YSCAL
     + /150,99,80,60,60,1500,4500,122*0./
      DATA LT/100*0/
      DATA L11/0,4,0,4,0,0,4,1,0,0/
      DATA L1220/90*0/
      DATA L21/24,2,0,7,0,3,2,2,1,0/
      DATA L22/32,8,0,7,0,4,4,2,0,0/
      DATA L23/10*0/
      DATA L24/24,3 0,5,0,3,2,2,1,0/
      DATA L25/10*0/
      DATA L2630/50*0/
      DATA L31/7,6,0,3,0,1,2,2,1,0/
      DATA L32/6,5,0,4,0,0,3,3,1,0/
      DATA L33/10,20,0,3,0 1,2,2,1,0/
      DATA L34/20,20,0,3,0,2,2,2,1,0/
      DATA L3540/60*0/
      DATA L41/4,3,0,4,0,1,2,2,1,0/
      DATA L42/10*0/
      DATA L4350/80*0/
      DATA L51/37,43,0,10,0,5,3,3,1,0/
      DATA L52/124,24,0,10,0,6,4,4,1,0/
      DATA L53/37,43,0,10,0,5,3,3,1,0/
```

BLOCK DATA (Continued)

```
      DATA L54/37,43,0,10,0,5,3,3,1,0/
      DATA L55/10*0/
      DATA L56/10,10,0,5,0,1,3,2,0,0/
      DATA L57/20,30,0,5,0,1,1,1,0,0/
      DATA L5860/30*0/
      DATA L61/2,0,0,12,0,1,10,1,0,0/
      DATA L62/6,10,0,7,0,2,5,2,0,0/
      DATA L63/18,18,0,9,0,2,7,2,0,0/
      DATA L6470/70*0/
      DATA L71/5,11,0,2,0,1,2,1,0,0/
      DATA L72/10*0/
      DATA L73/10*0/
      DATA L7480/70*0/
      DATA L81/8,8,0,2,0,1,2,2,1,0/
      DATA L82/24,25,0,8,0,2,8,6,0,0/
      DATA L83/8,5,0,4,0,1,4,2,0,0/
      DATA L8490/70*0/
      DATA L91/10*0/
      DATA L9293/81,1,0,5,0,3,0,0,0,0,24,3,0,5,0,3,0,0,0,0/
      DATA L9495/10*0,112,15,0,7,0,2,0,0,0,0/
      DATA L9697/20*0/
      DATA L98/81,1,0,13,0,2,0,0,0,0/
      DATA L99100/0,50,0,12,16*0/
      DATA L101/32,20,0,12,0,2,2,2,1,0/
      DATA L102/12,15,63,32,0,2,2,2,1,0/
      DATA L103/40,100,0,12,0,2,6,2,0,0/
      DATA L104/5,13,0,5,0,1,2,2,1,0/
      DATA L105/40,100,0,12,0,2,6,2,1,0/
      DATA L106/0,25,0,15,0,0,6,6,1,0/
      DATA L107/2,56,0,15,0,1,6,6,1,0/
      DATA L108/48,40,0,14,0,2,4,2,0,0/
      DATA L109/10*0/
      DATA L110/10*0/
      DATA LEND/400*0/
      END
```

COMP SUBROUTINE

```
      SUBROUTINE COMP
C**** REVISED DECEMBER 1979 ****
      DOUBLE PRECISION DD
      COMMON NTELPL,NTOLPL,IPT,IPOINT,NPTS,INEL,KNEL,NTOPL,NLPLT(61,3),
     1 PQLEG(90,12),ND(150,10)
      COMMON/SUB/PARM(150,9),PM(1500),QM(1500),P(300),Q(300),C(300)
     1,Z(300),RHO(20),S2ORHO(20),VISC(20),BULK(20),TEMP(20),PVAP(20)
     2,ATPRES,T,DELT,TFINAL,PLTDEL,PI,TITLE(20),LEGN,ICON
     3,KTEMP(99),LSTART(150),NLPT(150),LTYPE(99),NC(99),INX,INZ
     4,INV,ISTEP,NLINE,NEL,IND,IENTR,MNLINE,MNEL,MNLEG,MNNODE,MNPLOT
     5,MNLPTS,MDS,YSCAL(61,2)
      COMMON/COMPD/D(4500),L(1500),LE(99,4)
      DIMENSION DD(1400),LT(10,150),LC(1)
      EQUIVALENCE (L(1),LT(1,1)),(D(1),DD(1)),(C(1),LC(1))
      DATA L1,L4,NX/1,1,2/
      IND=1
      IF(IENTR) 1000,2000,2000
 1000 CONTINUE
      DO 1001 I=1,10
      DO 1002 J=1,150
 1002 ND(J,I)=LT(I,J)
 1001 CONTINUE
 1003 CONTINUE
      NCI=0
C     THIS READ STATEMENT INPUTS THE FOLLOWING DATA
C     I     =INDIVIDUAL COMPONENT NUMBER
C     LTYPE =COMPONENT TYPE
      READ(5,170) I,LTYPE(I),LDATAC,(L(L4-1+J),J=1,12),KTEMP(I)
      WRITE(6,500) I,I,LTYPE(I),LDATAC,(L(L4-1+J),J=1,12),KTEMP(I)
      NDATAC=LDATAC/1000
      LDATAC=LDATAC-NDATAC*1000
      IF(KTEMP(I))11,12,13
   11 KTEMP(I)=10-KTEMP(I)
      GO TO 13
   12 KTEMP(I)=1
   13 CONTINUE
      NLIM=LDATAC*8
C***  LDATAC EQUALS THE NUMBER OF ELEMENT REAL DATA CARDS
      KTYPE=LTYPE(I)
      NC(I)=ND(KTYPE,7)
      IF(NC(I).EQ.0) GO TO 7
      KK=L4
      NN=L4+NC(I)-1
      DO 4 N=KK,NN
      IF(L(N)) 1,2,3
    1 L(N)=-L(N)*2-1
      LC(L(N))=0
      NCI=NCI+1
```

COMP (Continued)

```
      GO TO 4
    2 L(N)=MNLINE*2-ND(KTYPE,9)
      GO TO 4
    3 L(N)=L(N)*2
      LC(L(N))=0
      NC1=NC1+1
    4 CONTINUE
    5 IF(L(NN).LT.MNLINE*2-1) GO TO 6
      NC(I)=NC(I)-1
      NN=NN-1
      GO TO 5
    6 CONTINUE
      INZ=NCI
      IF(NC1.NE.NC(I).AND.ND(KTYPE,10).NE.0)GO TO 420
      IF(NC(I).LT.ND(KTYPE,8)) NC(I)=ND(KTYPE,8)
    7 N1=ND(KTYPE,1)
      N2=ND(KTYPE,2)
      N3=ND(KTYPE,3)
      N4=ND(KTYPE,4)
      IF(NLIM.EQ.0) GO TO 15
      READ(5,450) (D(L1+N-1),N=1,NLIM)
      IF(IND.NE.I) GO TO 410
      NN=L1
      DO 10 KK=1,LDATAC
      WRITE(6,510)KK,(D(NN+N-1),N=1,8)
      NN=NN+8
   10 CONTINUE
   15 INX=0
      IF (NDATAC.NE.0) GO TO 20
      IF (LDATAC.GT.ND(KTYPE,6)) N1=NLIM
      LE(I,1)=L1
      LE(I,2)=L1+N1
      GO TO 30
   20 CONTINUE
      LE(I,1)=LE(NDATAC,1)
      LE(I,2)=L1
      INX=1
   30 IF(NX*N3.LE.1) GO TO 40
      LE(I,3)=(LE(I,2)+N2+3)/2
      L1=(LE(I,3)+N3)*2+1
      GO TO 50
   40 LE(I,3)=LE(I,2)+N2
      L1=LE(I,3)+N3
   50 LE(I,4)=L4
      L4=L4+N4
      ENTRY COMPE
 2000 CONTINUE
      KTYPE=LTYPE(IND)
      NTYPE =KTYPE/10
      N1=LE(IND,1)
      N2=LE(IND,2)
      N3=LE(IND,3)
      N4=LE(IND,4)
      GO TO (210,220,230,240,250,260,270,280,290,300),NTYPE
```

```
C 360 GO TO 400
  210 CONTINUE
      CALL BRAN11 (D(N1),D(N2),DD(N3),L(N4))
      GO TO  400
  220 CONTINUE
      KTYPE=KTYPE-20
      GO TO (221,222,400,224) KTYPE
  221 CALL VALV21 (D(N1),D(N2),DD(N3),L(N4))
      GO TO 400
  222 CALL VALV22 (D(N1),D(N2),DD(N3),L(N4))
      GO TO 400
  224 CALL VALV24(D(N1),D(N2),DD(N3),L(N4))
      GO TO 400
  230 CONTINUE
      KTYPE=KTYPE-30
      GO TO (231,232,233,234),KTYPE
  231 CALL CVAL31 (D(N1),D(N2),DD(N3),L(N4))
      GO TO 400
  232 CALL CREL32 (D(N1),D(N2) DD(N3),L(N4))
      GO TO 400
  233 CALL CVAL33(D(N1),D(N2),DD(N3),L(N4))
      GO TO 400
  234 CALL CVAL34(D(N1),D(N2),DD(N3),L(N4))
      GO TO 400
  240 CONTINUE
      CALL REST41 (D(N1),D(N2),DD(N3),L(N4))
      GO TO 400
  250 CONTINUE
      KTYPE=KTYPE-50
      GO TO (251,252,253,254,400,256 257),KTYPE
  251 CALL PUMP51 (D(N1),D(N2),DD(N3),L(N4))
      GO TO 400
  252 CALL PUMP52 (D(N1),D(N2),DD(N3),L(N4))
      GO TO 400
  253 CALL PUMP53 (D(N1),D(N2),DD(N3),L(N4))
      GO TO 400
  254 CALL PUMP54(D(N1),D(N2),DD(N3),L(N4))
      GO TO 400
  256 CALL MTR56(D(N1),D(N2),DD(N3),L(N4))
      GO TO 400
  257 CALL PUMP57(D(N1),D(N2),DD(N3),L(N4))
      GO TO 400
  260 CONTINUE
      KTYPE=KTYPE-60
      GO TO (261,262,263),KTYPE
 261  CALL RSVR61 (D(N1),D(N2),DD(N3),L(N4))
      GO TO 400
  262 CALL RSVR62 (D(N1),D(N2),DD(N3),L(N4))
      GO TO 400
  263 CALL RSVR63 (D(N1),D(N2),DD(N3),L(N4))
      GO TO 400
  270 CONTINUE
      KTYPE=KTYPE-70
      GO TO (271),KTYPE
```

```
271 CALL ACUM71(D(N1),D(N2),DD(N3),L(N4))
    GO TO 400
280 CONTINUE
    KTYPE=KTYPE-80
    GO TO (281,282,283),KTYPE
281 CALL FILT81 (D(N1),D(N2),DD(N3),L(N4))
    GO TO 400
282 CALL FILT82 (D(N1),D(N2),DD(N3),L(N4))
    GO TO 400
283 CALL FILT83(D(N1),D(N2),DD(N3),L(N4))
    GO TO 400
290 CONTINUE
    KTYPE=KTYPE-90
    GO TO (400,292,293,400,295,400,400,298,299),KTYPE
292 CALL CAD92 (D(N1),D(N2),DD(N3),L(N4))
    GO TO 400
293 CALL CAD93 (D(N1),D(N2),DD(N3),L(N4))
    GO TO 400
295 CALL APU95 (D(N1),D(N2),DD(N3),L(N4))
    GO TO 400
298 CALL CAD98 (D(N1),D(N2),DD(N3),L(N4))
    GO TO 400
299 CALL CAD99 (D(N1),D(N2),DD(N3),L(N4))
    GO TO 400
300 CONTINUE
    KTYPE=KTYPE-100
    GO TO (301,302,303,304,305,306,307,308),KTYPE
301 CALL ACT101 (D(N1),D(N2),DD(N3),L(N4))
    GO TO 400
302 CALL ACT102 (D(N1),D(N2),DD(N3),L(N4))
    GO TO 400
303 CALL ACT103 (D(N1),D(N2),DD(N3),L(N4))
    GO TO 400
304 CALL ACT104 (D(N1),D(N2),DD(N3),L(N4))
    GO TO 400
305 CALL ACT105 (D(N1),D(N2),DD(N3),L(N4))
    GO TO 400
306 CALL ACT106 (D(N1),D(N2),DD(N3),L(N4))
    GO TO 400
307 CALL ACT107 (D(N1),D(N2),DD(N3),L(N4))
    GO TO 400
308 CALL ACT108 (D(N1),D(N2),DD(N3),L(N4))
400 CONTINUE
    IF(INEL.NE.0.OR.IND.GE.NEL) RETURN
    IND=IND+1
    IF(IENTR) 1003,2000,2000
410 WRITE (6,180)
    WRITE(6,999)
999 FORMAT(10X,31HPROGRAM STOP IN SUBROUTINE COMP)
    STOP 6001
420 WRITE(6,190)IND
190 FORMAT(5X,42HTHERE ARE MISSING CONNECTIONS IN COMP NO    ,I5)
    WRITE(6,999)
    STOP 6001
```

COMP (Continued)

```
170 FORMAT (16I5)
180 FORMAT ( 35H THE ELEMENT CARDS ARE OUT OF ORDER  )
450 FORMAT (8E10.0)
500 FORMAT(1H0,5X,6HCOMP#,I5,2X,12HINTEGER DATA,2X,16I5,)
510 FORMAT(/,5X,16HREAL DATA CARD # ,I5,2X,8E12.4)
    END
```

APPENDIX P
LINE DATA
HYTRAN USER MANUAL (AFAPL-TR-76-43, VOL. I)

5.0 LINE DATA

The number of cards used in this group is equal to the number of lines entered on Card 3, and though they can be stacked in any order within the group, it is advised that the numerical order be used. An error message will be written when this condition is encountered but the program will continue. A line number may not be omitted or used twice. An error message will also be written if this condition is encountered, however the run will not stop.

A type # is used to differentiate between hard lines and hoses. As noted in Section 3.0, one line number can be used to represent any number of lines in series provided the diameter wall thickness and modulus of elasticity (effective bulk modulus if a hose) of each line are identical.

Two or more lines with different parameters may be joined together without using a branch or other component as a connection. These lines must be numbered consecutively, otherwise a 6002 error will be written and the run will stop.

Dead ended lines must have a 10 written in the type column of the line data card. Dead ended lines must indicate flow away for the component it is connected to.

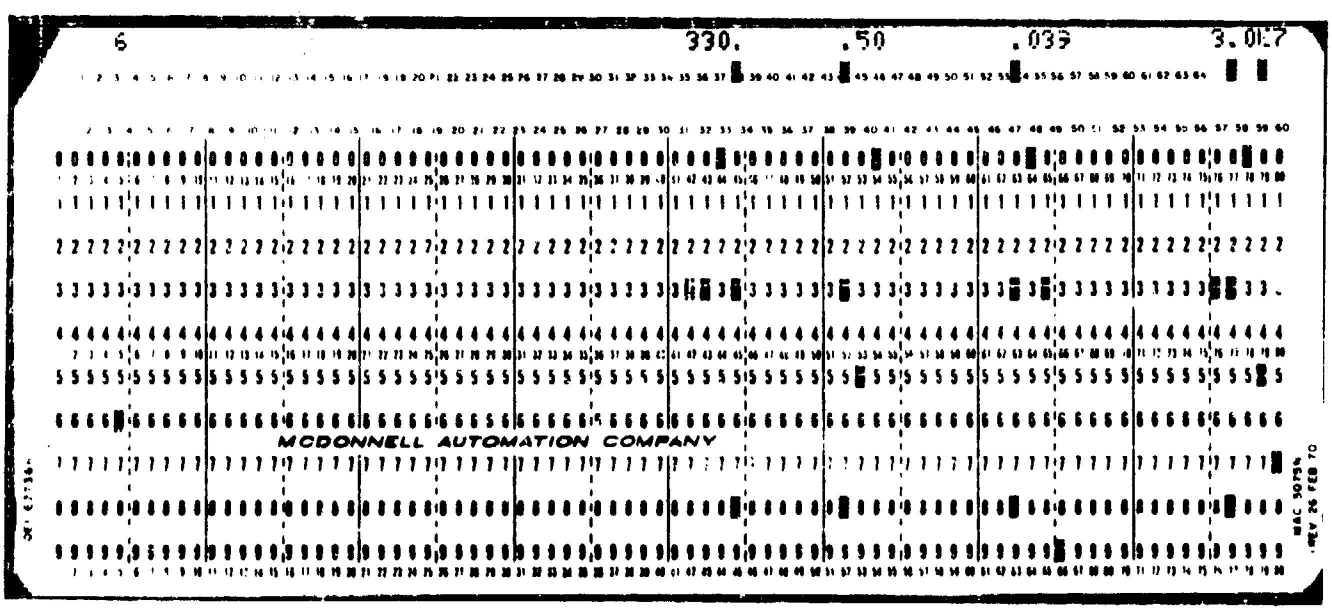
5.1 RIGID LINES

Type number zero is a rigid line. The majority of aircraft lines will fall under this category. True bend angles less than 90° are summed and input in columns 26 through 30. Angles equal to or greater than 90° are summed and input in columns 31 through 35.

CARD NUMBER 1

COLUMN	FORMAT	DATA	DIMENSIONS
1-5	I5	Line Number	-
6-10	I5	Type number = 0 or 10 for dead ended line, 2 for lossless line	-
11-15	I5	Percentage Increase in Fitting Friction	-
16-20	I5	Number of 45° Elbows	-
21-25	I5	Number of 90° Elbows	-
26-30	I5	Total of Bend Angles Less than 90°	deg
31-35	I5	Total of Bend Angles Greater Than or Equal to 90°	deg
36-40	I5	Temperature/Pressure Code (See Page 4.0-2)	-
41-50	E10.0	Total Length Including Fittings	in
51-60	E10.0	Outside Diameter	in
61-70	E10.0	Wall Thickness	in
71-80	E10.0	Modulus of Elasticity	psi

EXAMPLE CARD



Card 3. This card inputs the total number of lines, the number of components, fluid type number, optional fluid parameters (viscosity, density, bulk modulus and vapor pressure). Note: If a vapor pressure is not input the program will use a value of 2 psia. The fluid type number selects the fluid data to be used from tabulated data stored in the program and adjusts the fluid properties to the maximum and minimum pressures. The program is set up to run with either of the following fluid types at any temperature from -65°F to 300°F:

Type #1	MIL-H-5606B
Type #2	MIL-H-83282
Type #3	Skydrol 500B

In addition, the user can input fluid data for any fluid (for the maximum temperature specified on Card 2) by using a Type #0 and by inputting viscosity, density, and bulk modulus, and vapor pressure. This fluid data is not 'pressure' adjusted and is used as input. Note: Columns 21 through 50 can be left blank if fluid type #1, #2, or #3 are used.

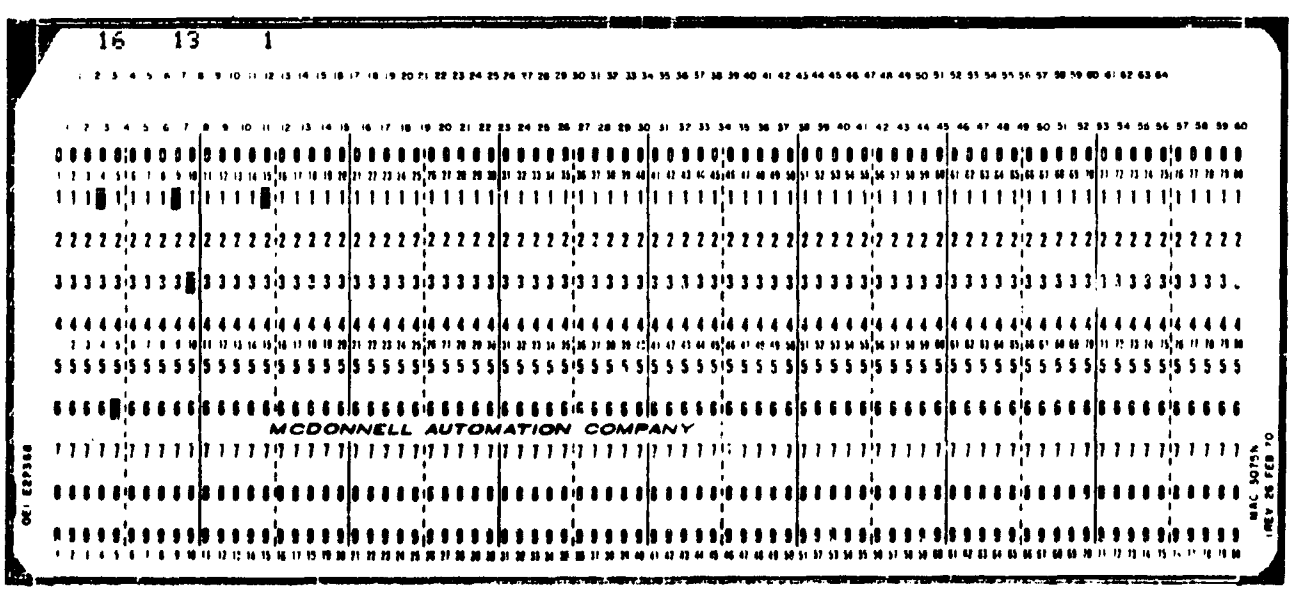
Two or more lines with different parameters may be joined together without using a branch or other component as the connection. These lines must be numbered sequentially, otherwise a 6002 error will be written and the run will stop.

Dead ended lines must have a 10 written in the type column of the line data card.

CARD NUMBER 3

COLUMN	FORMAT	DATA	DIMENSIONS
1-5	I5	Total Number of Lines	-
6-10	I5	Total Number of Components	-
11-15	I5	Fluid Type Number	-
16-20	I5	This value is set to 0, +N, -N to depressurize a system. See Page 6.54-2 for a description of this function.	-
21-30	E10.0	Fluid Viscosity	in/sec
31-40	E10.0	Fluid Density	lb-sec^2/in^4
41-50	E10.0	Fluid Adiabatic Bulk Modulus	psi
51-60	E10.0	Vapor Pressure	psia
61-80	E10.0	Not Used	-

EXAMPLE CARD



APPENDIX P
SUBROUTINE LINE
HYTRAN TECHNICAL MANUAL (AFAPL-TR-76-43, VOL. II)

5.1 SUBROUTINE LINE

LINE divides each line into segments, the length of each segment being greater than or equal to the velocity of sound in fluid multiplied by the time step, DELT. LINE then calculates the flows and pressures for all the interior line points as well as the end point characteristics. These characteristics are subsequently used by component subroutines to calculate pressures and flows at the ends of each line where the lines and components interface.

5.1.1 Math Model

The equation of motion and the continuity equation, describing the one-dimensional unsteady flow of a compressible fluid in an elastic fluid line, equations (1) and (2), involve the actual distributed parameters and include the nonlinear viscous terms. These partial differential equations are transformed into total differential equations by use of the method of characteristics. A finite difference method is then used to place the total differential equations in a form suitable for numerical solution on a digital computer.

Basic Equations

The equation of motion is:

$$\frac{\partial P}{\partial X} + \frac{RHO}{A} * \frac{\partial Q}{\partial t} + F(Q) = 0 \qquad (1)$$

The condition of continuity yields:

$$\frac{\partial Q}{\partial X} + \frac{A}{RHO{*}\ a{*}{*}2} * \frac{\partial P}{\partial t} = 0 \qquad (2)$$

where	P = Pressure	psia
	Q = Flow	cis
	A = Flow area of line	in^2
	t = Time	sec

RHO = Fluid mass density		(lb-sec**2)/in.**4
F(Q) = Pressure loss as a function of flow		psi
a = speed of the wave propagation = (BULK/RHO)*1/2		in./sec
BULK = Bulk modulus of the fluid		psi
X = Distance along line		in.

Method of Characteristics

The basic equations (1) and (2) can be transformed into a pair of total differential equations, the validity of which is restricted to certain lines in the x-t plane called the characteristics. Integration along the characteristics, Fig. 5.1-1, yields the following algebraic equations.

For convenience let

$$Z = \frac{RHO*a}{A}$$

(the number, Z_c, is the characteristic impedance of a frictionless line) and let

$$\text{C left:} \quad C_L = P_{VW} - Z*Q_{VW} + \frac{a\,\Delta\,t}{2}\,[F(Q_{WT}) + F(Q_{VW})] \quad (3)$$

$$\text{C right:} \quad C_R = P_{WX} - Z*Q_{WX} + \frac{a\,\Delta\,t}{2}\,[F(Q_{WT}) + F(Q_{WX})] \quad (4)$$

then

$$P + ZQ_P - C_R = 0 \quad (5)$$

$$CR = P_{WT} - Z * Q_{WT}$$

$$P_P - Z_C Q_P - C_L = 0 \quad (6)$$

$$CL = P_{WT} + Z * Q_{WT}$$

$$P_{WT} = + \frac{C_R + C_L}{2} \quad (7)$$

$$Q_{WT} = \frac{C_L - C_R}{2*Z} \quad (8)$$

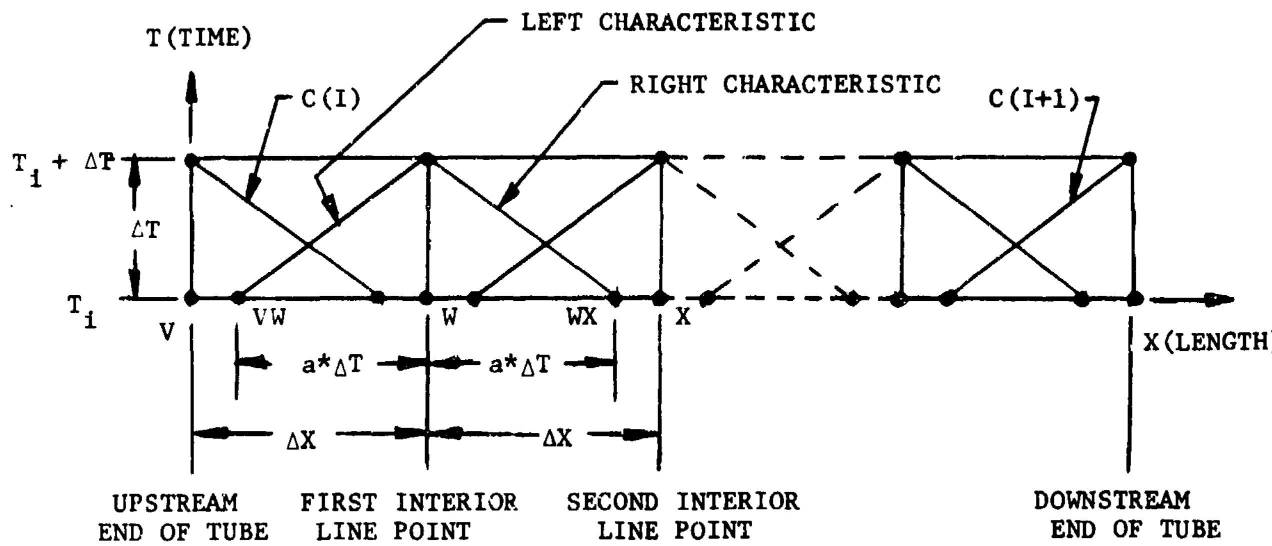


FIGURE 5.1-1 GRID OF CHARACTERISTICS WITH INTERPOLATION

Equations (7) and (8) apply only to the interior sections of the fluid line. Points at the boundary use only one of the characteristic equations to be solved simultaneously with the boundary condition. C_R and C_L are still functions of the unknown, Q_{WT}. Therefore, an iteration is used to calculate Q_{WT}. C_R and C_L are first determined using only a rectangular rule to approximate the friction.

$$C_L = P_{VW} + Z^*Q_{VW} - a^*\Delta t^*F(Q_{VW}) \qquad (3a)$$

$$C_R = P_{WX} - Z^*Q_{WX} + a^*\Delta t^*F(Q_{WX}) \qquad (4a)$$

This gives a very close approximation of Qp, which then can be improved by using the trapezoidal rule to evaluate C_R and C_L.

There are two ways to apply these equations to a line system: either one has to modify the wave speeds slightly to make $n = L/(a^*\Delta t)$ an integer in each individual line (L is the length of a line), or one has to use the interpolating grid of Fig. 5.1-1 which has been applied in HYTRAN and is discussed in more detail by Streeter and Wylie (5).

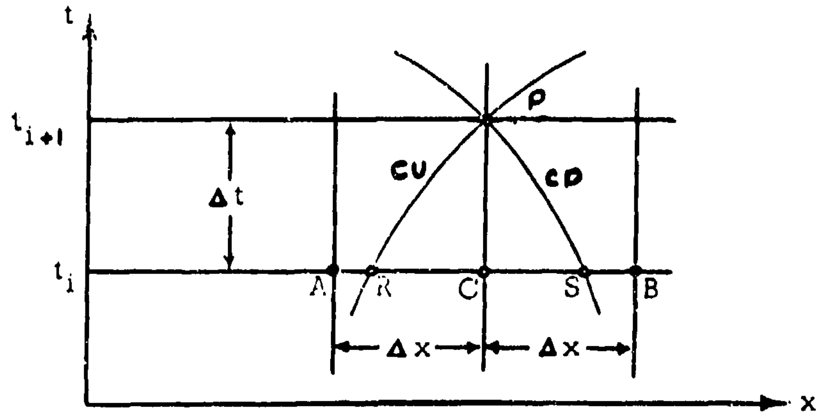


FIGURE 5.1-2 CHARACTERISTICS IN X,t PLANE

5.1.2 Assumptions

1. Transition from laminar to turbulent flow is assumed to occur at a Reynolds number of 1200.

2. Friction factors used are based on circular cross-section, smooth I.D., drawn tubing.

3. See Appendix A for assumptions associated with method of characteristics.

5.1.3 Computation Methods

Section 1000

The type of line N is isolated and the speed of sound and characteristic impedance are calculated

```
IF(LDN(N,1).EQ.1) GO to 70
A = 1.8*(PARM(N,3)+PARM(N,2)/2.0)*BULK(IT)
A = BULK(IT)/RHO(IT)*(1.0+A/(PARM(N,4)*PARM(N,3))))
GO to 75
70 BULKH = BULK(IT)*PARM(N,4)/(BULK(IT)+PARM(N,4))
A = BULKH/RHO(IT)
75 A=SQRT(A)
PARM(N,6) = RHO(IT)*A*4.0/(PI*PARM(N,2)**2)
```

If the speed of sound is not large enough to produce two line points, a fix-up is taken and the percent error in the adjusted speed of sound is printed

```
CONST = PARM(N,1)/DELT
IF(CONST.GE.A) GO to 78
PC = 100.-CONST*100./A
WRITE(6,300)N,PC
A = CONST
78 PARM(N,7) = A
NLPT(N) = CONST/A+1.01
```

DELX is then calculated

```
PARM(N,5) = PARM(N,1)/(NLPT(N)-1)
```

Equivalent line length of bends and fittings is calculated for line N

```
EQUIVL = PARM(N,2)*(LDN(N,3)*12+
         LDN(N,4)*57.0+LDN(N,5)/45.*
         4.65 (LDN(N,6)/90.+1.)*3.75
```

The equivalent line length is then added to the actual line length and is used to calculate the laminar and turbulent flow constants

```
PARM(N,8) = 128./PI*VISC(IT)*RHO(IT)/(DIA**4.)*
            (PARM(N,1)+EQUIVL)
PARM(N,9) = .213*RHO(IT)*(VISC**.25)/(DIA**4.75)
            *(PARM(N,1)+EQUIVL)
```

The laminar and turbulent flow constants are set to 1.0E-6 for a lossless line (LDN(N,1)+2). The above steps are repeated until the data for all lines are calculated. The data is then printed. A constant for subsequent interpolation calculations, transition flow and address of line points are calculated and stored for each line. Finally, variables used for plugged and open connections are initialized.

Dynamic friction entry DYFRI is called to calculate and store line constants for the dynamic pressure loss equation. The returned value of DYFRI is multiplied times two and stored in the DYF( ) array.

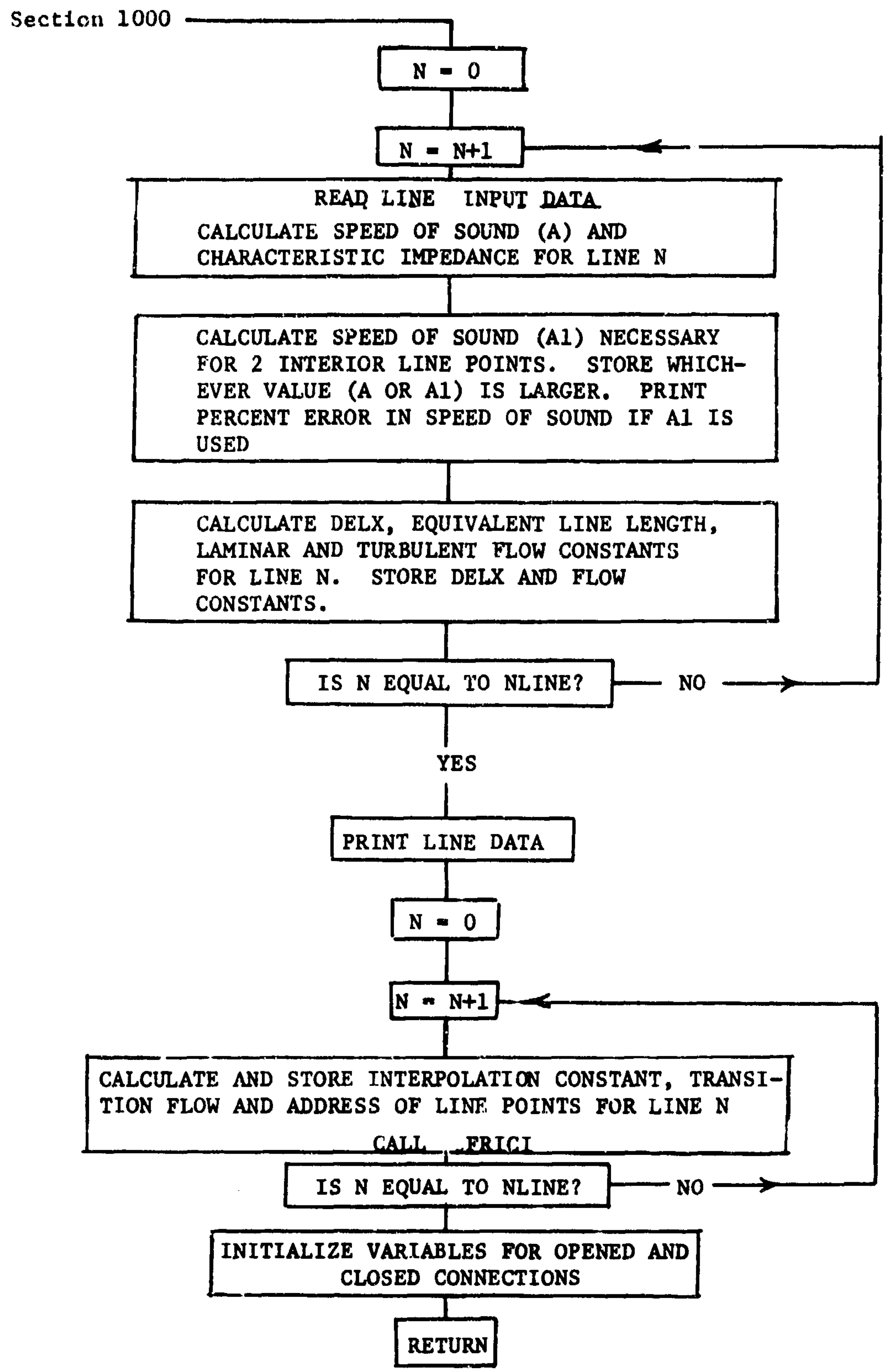


FIGURE 5.1-3 LINE SUBROUTINE FLOW DIAGRAM

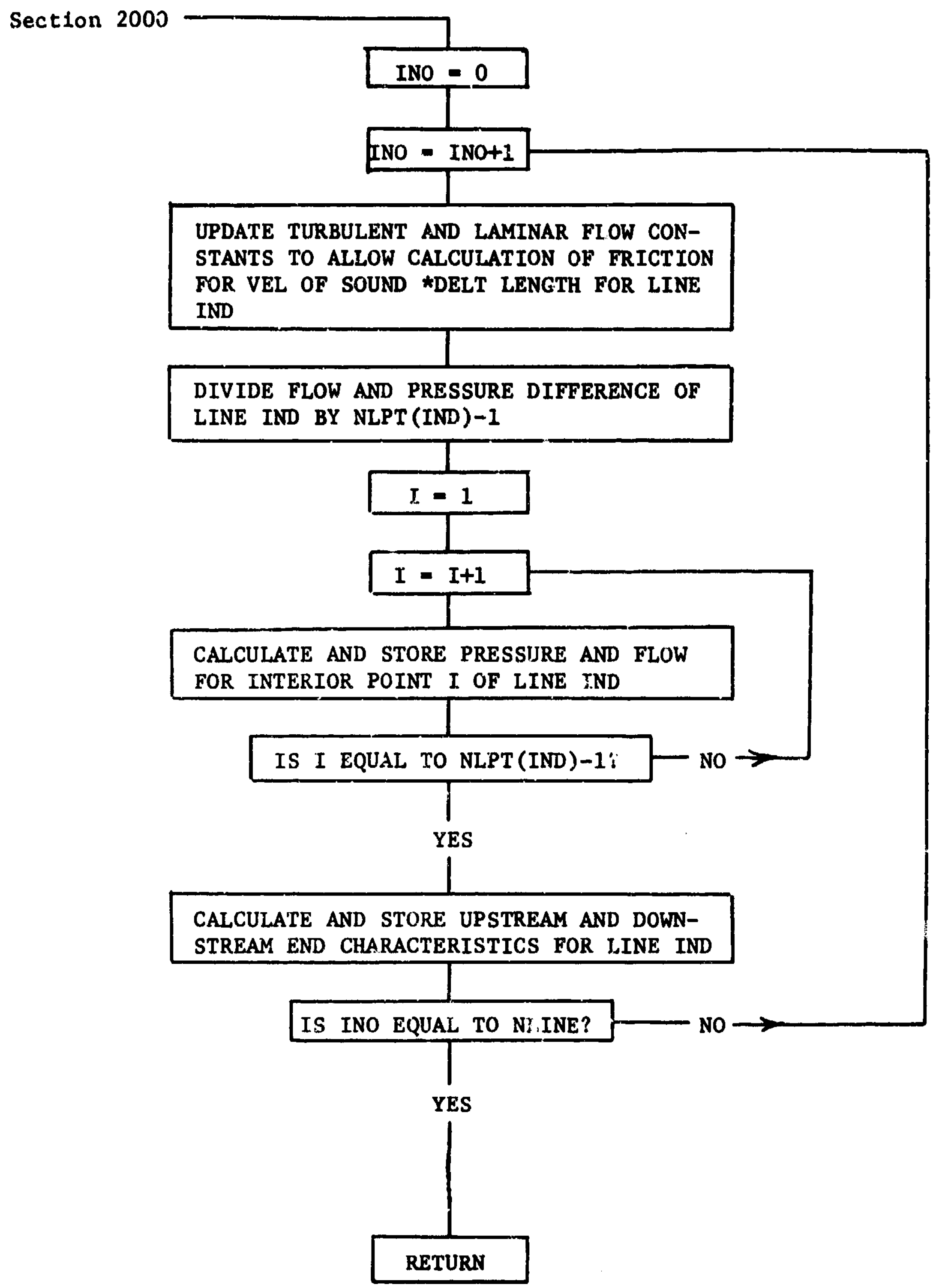


FIGURE 5.1-3 CONT.

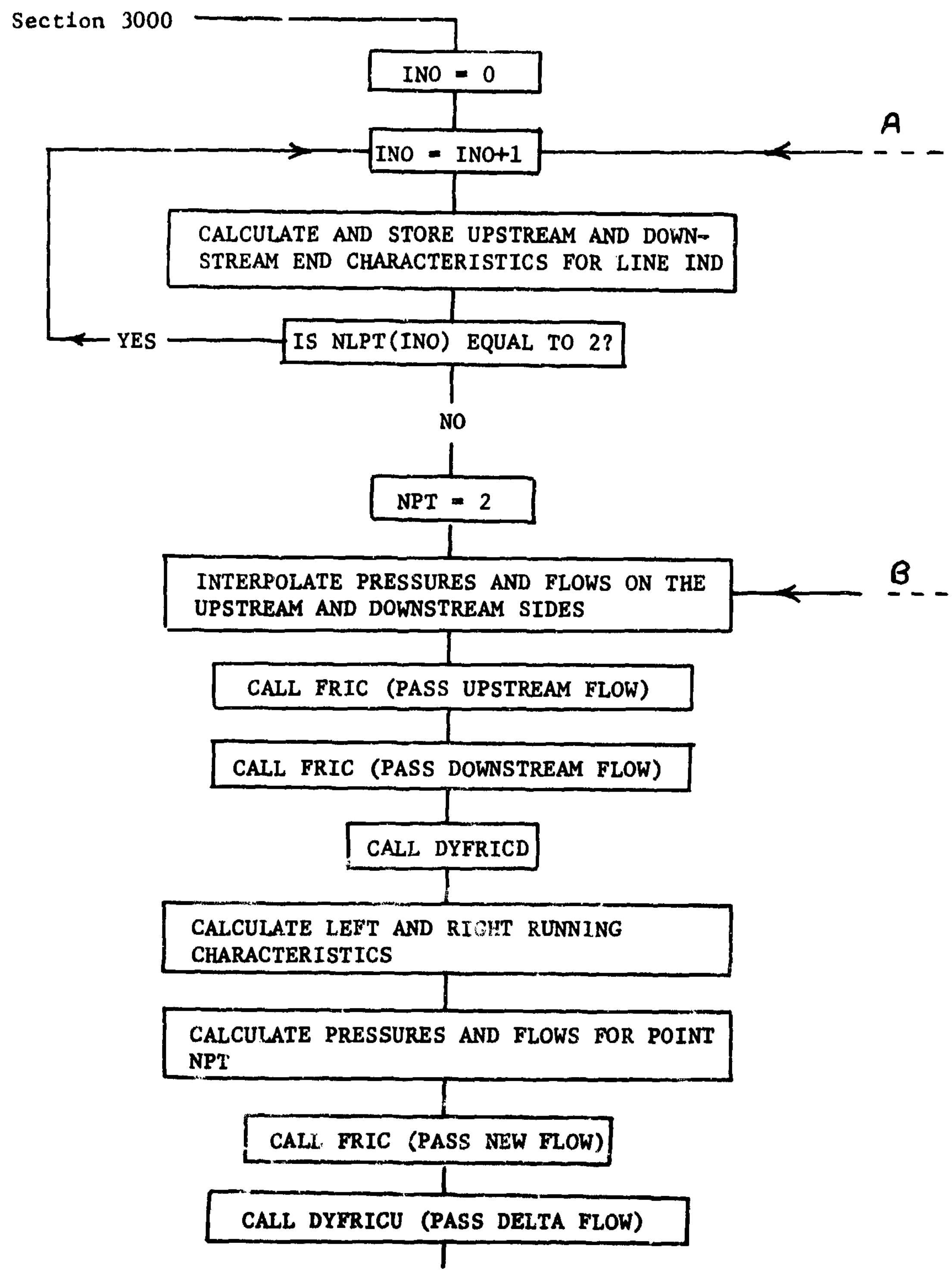


FIGURE 5.1-3 (CONT.

Section 3000 (Cont)

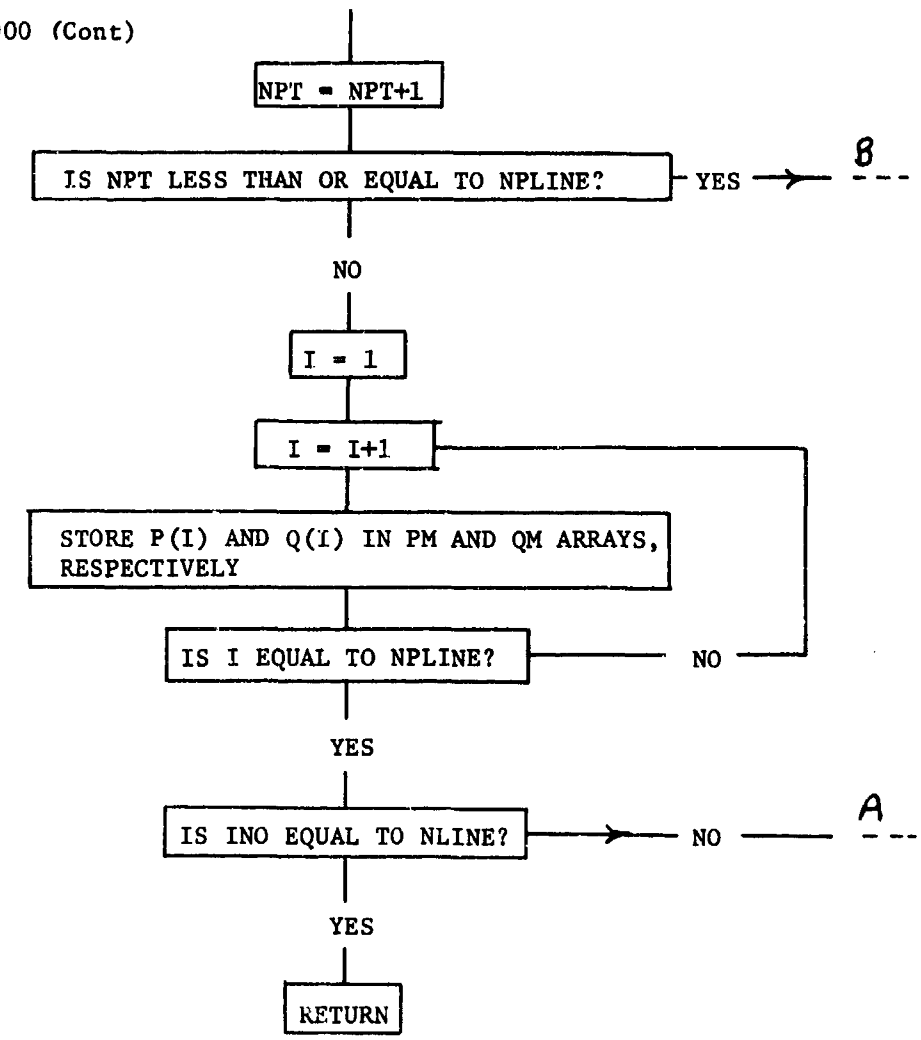


FIGURE 5.1-3 CONT.

Section 2000

The laminar and turbulent flow constants are updated to allow calculation of pressure drop for velocity of sound * DELT length of each line.

PARM(IND,8)=PARM(IND.8)*PARM(IND.7)*DELT/PARM(IND,1)

PARM(IND,9)=PARM(IND,9)*PARM(IND,7)*DELT/PARM(IND,1)

The pressure and flow differences for each line are divided by the number of line increments (number of line points -1).

DELP = (PPM1 - PPM2)/(N-1)

DELQ = (QQM1 - QQM2)/(N-1)

The pressures and flows are next calculated and stored for each interior line point. Pressures and flows on the upstream and downstream sides are interpolated then upstream and downstream end point characteristics are calculated and stored for all lines

C(LUP) = PPI-ZC*QPI+FRIC(QPI)

C(LDWN) = PMI+ZC*QMI-FRIC(QMI)

Section 3000

A DO loop is used to calculate the values for each line.

DO 140 IND = 1, NLINE

Pressures and flows on the upstream and downstream sides are interpolated.

PPI = PM(M)-PARM(IND,3)*(PM(M)-PM(M+1))

QPI = QM(M)-PARM(IND,3)*(QM(M)-QM(M+1))

PMI = PM(M+N-1)-PARM(IND,3)*(PM(M+N-1)-PM(M+N-2))

QMI = QM(M+N-1)-PARM(IND,3)*(QM(M+N-1)-QM(M+N-2))

Upstream and downstream end point characteristics are calculated and stored.

```
C(IND*2-1) = PPI-ZC*QPI+FRIC(QPI)
C(IND*2) = PMI+ZC*QMI-FRIC(QMI)
```

If there are only 2 points for the given line, control then passes to the end of the DO loop.

```
IF (N.LE.2) GO to 140
```

If there are interior points, the flows and pressures at these points are then calculated. Pressures and flows on the upstream and downstream sides of interior point NPT are interpolated and steady state friction for each of these flows is calculated by calling FRIC.

```
FRICM = FRIC(QMI)
FRICP = FRIC(QPI)
```

The decayed dynamic friction value is calculated by calling DFRICD.

```
DYFRIC = DFRICD(QMI,MPT,JJ,JTEMP)
```

The characteristics are now calculated for interior point NPT.

```
CL = +PMI+ZC*QMI-FRICM-DYFRIC
CR = +PPI-ZC*QPI+FRICP+DYFRIC
```

The first approximation of the pressure and flow at point NPT are calculated.

```
P (NPT) =  (CR+CL)/2.
Q (NPT) =  (CL-CR+QM(MPT)*DYF(IND))/(2.*Z6+DYF(IND))
```

Friction is recalculated using the new flow.

```
FRICR=FRIC (Q(NPT))
```

Dynamic friction is updated using the difference between new and old flows.

```
DYFRIC=DFRICU(DELQ,MPT,JTEMP)
```

If a line segment is cavitated, the dynamic friction terms and the updated terms are set to zero until pressure returns to the line.

Once the pressures and flows for all points have been calculated, they are stored in the PM and QM arrays.

```
DO 130 I = 2, NPLINE
PM(M+I-1) = P(I)
QM(M+I-1) = Q(I)
130 CONTINUE
```

5.1.4 Approximations

See Section III, Paragraph 4 of AFWAL-TR-80-2039

5.1.5 Limitations

See Section III Paragraph 4 of AFWAL-TR-80-2039

5.1.6 Variable Names

A	Velocity of Sound in Fluid	IN/SEC
BULKH	Equivalent Bulk Modulus of Hose	PSI
CL	Left Running Characteristic	PSI
CONST	Constant-Line Length/DELT	IN/SEC
CR	Right Running Characteristic	PSI
DELP	Pressure Drop for DELX Length	PSI

DELQ	Flow Difference for DELX Length	CIS
DIA	Line I.D.	IN
DYF( )	Dynamic Friction Constant Array	-
DYFRIC	Dynamic Friction	PSI
EQUIVL	Equivalent Length of Line	IN
FRICM	Pressure Drop for DELX Length Using Flow QMI	PSI
FRICP	Pressure Drop for DELX Length Using Flow QPI	PSI
FRICR	Pressure Drop for DELX Length Using Flow Q (NPT)	PSI
I1	Fluid Temp/Pressure Number	--
INO	Counter	--
JJ	LSTART(IND)+NLPT(IND)-1	--
JTEMP	LINE CAVITATION INDICATOR	--
LDWN	Downstream Line End Address	--
LLPT	NLPT(IND)-1	--
LUP	Upstream Line End Address	--
M	Dummy Variable	--
MPM1	Address of Current Upstream Line Point	--
MPP1	Address of Current Downstream Line Point	--
MPT	Address LSTART(IND)+1	--
N	Counter	--
NPLINE	NLPT(IND)-1	--
NPT	Current Interior Line Point	PSI
PC	Percent Error	--
PLAST	Past Pressure Value	PSI
PMI	Interpolated Pressure on the Upstream Side	PSI
PPI	Interpoloated Pressure on the Downstream Side	PSI
PPM1	Dummy Variable	--
PPM2	Dummy Variable	--

QLAST	Past Flow Value	--
QMI	Interpolated Flow on the Upstream Side	CIS
QPI	Interpolated Flow on the Downstream Side	CIS
QQM1	Dummy Variable	--
QQM2	Dummy Variable	--
REN	Reynolds Number	--
ZC	Dummy Variable	--

5.1.7 Subroutine Listing

```
      SUBROUTINE LINE
      COMMON NTELPL,NTOLPL,IPT,IPOINT,NPTS,INEL,KNEL,NTOPL,NLPLT(61,3),
     1 PQLEG(90,12),LCS(90,10),ILEG(1400),PN(90),QN(90)
      COMMON/SUB/PARM(150,9),PM(1500),QM(1500),P(300),Q(300),C(300)
     1,Z(300),RHO(20),S2ORHO(20),VISC(20),BULK(20),TEMP(20),PVAP(20)
     2,ATPRES,T,DELT,TFINAL,PLTDEL,PI,TITLE(20),LEGN,ICON
     3,KTEMP(99),LSTART(150),NLPT(150),LTYPE(99),NC(99),INX,INZ
     4,INV,ISTEP,NLINE,NEL,IND,IENTR,MNLINE,MNEL,MNLEG,MNNODE,MNPLOT
     5,MNLPTS,MDS
C     THIS SUBROUTINE SIMULATES LINES AS DISTRIBUTED PARAMETER SYSTEMS
C     USING THE METHOD OF CHARACTERISTICS
      DIMENSION LDN(150,7),LC(300),DYF(150),LTEMP(150)
      EQUIVALENCE(ILEG(1),LDN(1)),(C(1),LC(1))
      IF(IENTR)1000,2000,3000
 1000 CONTINUE
C
C     THIS RO   DS LINE DASH NUMBERS(LDN) AND LINE PARAMETERS(PARM)-SEE
C     N       INDIVIDUAL LINE NUMBER
C     LDN(.,1)=LINE TYPE
C     LDN(.,2)=PERCENTAGE INCREASE IN LINE LOSS
C     LDN(.,3)=NUMBER OF 45 DEG ELBOWS
C     LDN(.,4)=NUMBER OF 90 DEG ELBOWS
C     LDN(.,5)=TOTAL OF BEND ANGLES .LT. 90 DEG   DEG
C     LDN(.,6)=TOTAL OF BEND ANGLES .GE. 90 DEG   DEG
C     LDN(.,7)= FLUID TEMPERATURE/PRESSURE NUMBER
C***  PARM(.,1)=LENGTH
C     PARM(.,2)=LINE OD   IN.
C     PARM(.,3)=WALL THICKNESS/CONSTANT
C     PARM(.,4)=MODULUS OF ELASTICITY   PSI/TRANSITION FLOW FOR GIVEN
C               REN,DIA + DENSITY   IN**3/SEC
      DO 1100 I=1,NLINE
      READ(5,1300) N,(LDN(N,J),J=1,7),(PARM(N,J),J=1,4)
 1300 FORMAT ( 8I5,4E10.0)
      IF(LDN(N,7))10,20,30
   10 LDN(N,7)=10-LDN(N,7)
      GO TO 30
   20 LDN(N,7)=1
   30 CONTINUE
      LTEMP(N)=LDN(N,7)
      IF(I.NE.N) WRITE(6,430) N
  430 FORMAT ( 43H  THE LINE CARDS ARE OUT OF ORDER AT NUMBER , I5)
 1100 CONTINUE
```

```
C      REN   =REYNOLD*S NUMBER
C      EQUIVL=EQUIVALENT LENGTH
C***   PARM(.,5)=DELX
C***   PARM(.,6)=CHARACTERISTIC IMPEDANCE
C***   PARM(.,7)=VELOCITY OF SOUND IN THE LINE
C      PARM(.,8)=LAMINAR FLOW CONSTANT FOR GIVEN DIA.VISC + DENSITY
C      PARM(.,9)=TURBULENT FLOW CONSTANT FOR GIVEN DIA,VISC + DENSITY
C      PARM(.,2) IS INPUTED AS LINE OD AND CONVERTED TO LINE ID
       REN=1200
       N=1
       DO 80 N=1,NLINE
       IT=LDN(N,7)
       M=N*2
       LC(M)=1
       LC(M-1)=1
       IF(LDN(N,1).LT.10) GO TO 65
       LC(M)=-1
       LDN(N,1)=LDN(N,1)-10
  65   IF(LDN(N,1).EQ.1) GO TO 70
       PARM(N,2)=PARM(N,2)-PARM(N,3)*2.0
       A=1.8*(PARM(N,3)+PARM(N,2)/2.0)*BULK(IT)
       A=BULK(IT)/(RHO(IT)*(1.0+A/(PARM(N,4)*PARM(N,3))))
       IF(LDN(N,1).EQ.2)A=BULK(IT)/RHO(IT)
       GO TO 75
  70   BULKH=BULK(IT)*PARM(N,4)/(BULK(IT)+PARM(N,4))
       A=BULKH/RHO(IT)
  75   A=SQRT(A)
       PARM(N,6)=RHO(IT)*A*4.0/(PI*PARM(N,2)**2)
       CONST=PARM(N,1)/DELT
       IF(CONST.GE.A) GO TO 78
       PC=100.-CONST*100./A
       WRITE(6,300)N,PC
       A=CONST
  78   PARM(N,7)=A
       NLPT(N)=CONST/A+1.01
       PARM(N,5)=PARM(N,1)/(NLPT(N)-1)
       IF(LDN(N,1).EQ.2)GO TO 79
       JJ=1
       IF(LDN(N,6).EQ.0)JJ=0
       EQUIVL=PARM(N,2)*(LDN(N,3)*12+LDN(N,4)*57.0+LDN(N,5)/45.*4.65+
      1       (LDN(N,6)/90.+1.)*3.75*JJ)+PARM(N,1)*LDN(N,2)/100
       DIA=PARM(N,2)
       PARM(N,8)=128./PI*VISC(IT)*RHO(IT)/(DIA**4.)*(PARM(N,1)+EQUIVL)
       PARM(N,9)=.213*RHO(IT)*(VISC(IT)**.25)/(DIA**4.75)*(PARM(N,1)
      1+EQUIVL)
       GO TO 80
  79   PARM(N,8)=1.0E-6
       PARM(N,9)=1.0E-6
  80   CONTINUE
```

```
      WRITE(6,340)
      WRITE(6,350)((I),(PARM(I,J),J=1,7),I=1,NLINE)
      M=-1
      LSTART(1)=1
      DO 90 N=1,NLINE
      IT=LDN(N,7)
      PARM(N,3)=PARM(N,7)*DELT/PARM(N,5)
      PARM(N,4)=.7854*REN*PARM(N,2)*VISC(IT)
      M=M+2
      IF(LSTART(N)+NLPT(N).GT.MNLPTS-2) GO TO 252
      LSTART(N+1)=NLPT(N)+LSTART(N)
      DYF(N)=DYFRI(DELQ,IT,I,N)*2.
      IF(LDN(N,1).EQ.2)DYF(N)=0.0
      Z(M)=PARM(N,6)
      Z(M+1)=Z(M)
   90 CONTINUE
C***
C     INITIALIZE VARIABLES USED FOR PLUGGED AND OPEN
C     CONNECTIONS, (MNLINE)=PLUGGED, (MNLINE-1)=OPEN
      LSTART(MNLINE)=MNLPTS-1
      NLPT(MNLINE)=2
      N=MNLINE*2
      C(N)=0.0
      C(N-1)=ATPRES
      Z(N)=1.0E25
      Z(N-1)=1.0E-10
C***  JUNK WILL BE STORED IN PM(MNLPTS-4) THRO PM(MNLPTS-1)
C***  LIKEWISE FOR QM()
      IF(NLINE.GT.MNLINE-1) GO TO 252
      RETURN
  252 WRITE(6,475) NLINE,LSTART(NLINE),NLPT(NLINE)
      WRITE(6,999)
  999 FORMAT(10X,31HPROGRAM STOP IN SUBROUTINE LINE)
      STOP 5101
  475 FORMAT(5X,9HERROR #5  ,3I10)
 2000 CONTINUE
      DO 2020 INO=1,NLINE
      IND=INO
      PARM(IND,8)=PARM(IND,8)*PARM(IND,7)*DELT/PARM(IND,1)
      PARM(IND,9)=PARM(IND,9)*PARM(IND,7)*DELT/PARM(IND,1)
      ZC=PARM(IND,6)
      N=NLPT(IND)
      M=LSTART(IND)
      PPM1=PM(M)
      PPM2=PM(M-1+N)
      DELP=(PPM1-PPM2)/(N-1)
      QQM1=QM(M)
      QQM2=QM(M-1+N)
      DELQ=(QQM1-QQM2)/(N-1)
      LLPT=N-1
```

```
      DO 2010 I=2,LLPT
      PPM1=PPM1-DELP
      QQM1=QQM1-DELQ
      PM(M-1+I)=PPM1
      QM(M-1+I)=QQM1
 2010 CONTINUE
      LUP=IND*2-1
      LDWN=IND*2
      P(LUP)=PM(M)
      Q(LUP)=-QM(M)
      P(LDWN)=PM(M+N-1)
      Q(LDWN)=QM(M+N-1)
C***
C     CALCULATE CR FOR UPSTREAM ENDPOINT
      PPI=PM(M)-PARM(IND,3)*(PM(M)-PM(M+1))
      QPI=QM(M)-PARM(IND,3)*(QM(M)-QM(M+1))
      PMI=PM(M+N-1)-PARM(IND,3)*(PM(M+N-1)-PM(M+N-2))
      QMI=QM(M+N-1)-PARM(IND,3)*(QM(M+N-1)-QM(M+N-2))
      C(LUP)=PPI-ZC*QPI+FRIC(QPI)
C     CALCULATE CL FOR DOWNSTREAM END POINT
      C(LDWN)=PMI+ZC*QMI-FRIC(QMI)
 2020 CONTINUE
      RETURN
 3000 CONTINUE
      DO 140 INO=1,NLINE
      IND=INO
      ZC=PARM(IND,6)
      N=NLPT(IND)
      M=LSTART(IND)
      JJ=M+N-1
      JTEMP=JJ
      NPLINE=N-1
      IF(PM(M).LE.PVAP(LTEMP(IND)).OR.PM(JJ).LE.PVAP(LTEMP(IND))
     +)JTEMP=-1
C     CALCULATE CR FOR UPSTREAM ENDPOINT
      PPI=PM(M)-PARM(IND,3)*(PM(M)-PM(M+1))
      QPI=QM(M)-PARM(IND,3)*(QM(M)-QM(M+1))
      PMI=PM(M+N-1)-PARM(IND,3)*(PM(M+N-1)-PM(M+N-2))
      QMI=QM(M+N-1)-PARM(IND,3)*(QM(M+N-1)-QM(M+N-2))
      C(IND*2-1)=PPI-ZC*QPI+FRIC(QPI)
C     CALCULATE CL FOR DOWNSTREAM END POINT
      C(IND*2)=PMI+ZC*QMI-FRIC(QMI)
      IF(N.LE.2) GO TO 140
```

```
C
C     ALGORITHM FOR INTERIOR POINTS OF LINES
      NPT=2
      MPT =M+1
  100 CONTINUE
      MPP1=MPT+1
      MPM1=MPT-1
C     INTERPOLATION ROUTINE
C     PPI - INTERPOLATED PRESSURE ON THE PLUS(DOWNSTREAM) SIDE
C     PMI - INTERPOLATED PRESSURE ON THE MINUS(UPSTREAM) SIDE
C     SIMILAR NOTATION IS USED FOR FLOWS
      QPI=QM(MPT)-PARM(IND,3)*(QM(MPT)-QM(MPP1))
      PPI=PM(MPT)-PARM(IND,3)*(PM(MPT)-PM(MPP1))
      PMI=PM(MPT)-PARM(IND,3)*(PM(MPT)-PM(MPM1))
      QMI=QM(MPT)-PARM(IND,3)*(QM(MPT)-QM(MPM1))
C     FIRST APPROXIMATE CL + CR
      FRICM=FRIC(QMI)
      FRICP=FRIC(QPI)
      DYFRIC=DFRICD(QMI,MPT,JJ,JTEMP)
      IF(FRICM.EQ.0.0)DYFRIC=0.0
      CL=PMI+ZC*QMI-FRICM-DYFRIC
      CR=PPI-ZC*QPI+FRICP+DYFRIC
C     CALCULATE PRESSURE(P) AND FLOW(Q) USING THE ABOVE APPROXIMATIONS
      P(NPT)= (CR+CL)/2.
      Q(NPT)=(CL-CR+QM(MPT)*DYF(IND))/(2.*ZC+DYF(IND))
      DELQ=Q(NPT)-QM(MPT)
      DYFRIC=DFRICU(DELQ,MPT,JTEMP,M)
      MPT =MPT+1
      NPT=NPT+1
      IF(NPT.LE.NPLINE) GO TO 100
C     AFTER FINDING P + Q FOR THE WHOLE LINE, PUT THESE VALUES INTO PM +
      DO 130 I=2,NPLINE
      PM(M+I-1)=P(I)
      QM(M+I-1)=Q(I)
  130 CONTINUE
  140 CONTINUE
  300 FORMAT(/,5X,44HFIX-UP TAKEN AT LINE 18,VEL OF SOUND IN LINE,I4,2X,
     12HIS,F8.1,17HPER CENT IN ERROR)
  340 FORMAT(//,10H LINE DATA,/,9H LINE NO.,8X,6HLENGTH,9X,8HINTERNAL
     1,7X,4HWALL,11X,10HMODULUS OF,5X,4HDELX,11X,14HCHARACTERISTIC,
     212H VELOCITY OF ,/,32X,
     3          3HDIA,12X,9HTHICKNESS,6X,10HELASTICITY,20X,9HIMPEDANCE
     4,11H      SOUND)
  350 FORMAT(/1X,I5,10X,F8.4,7X,F8.4,7X,F8.4,7X,  E10.3,5X,F8.4,7X,F8.4,
     16X,F11.4)
  360 FORMAT(1X,5X,I5,6E12.5)
      RETURN
      END
```

```
C     REN   =REYNOLD*S NUMBER
C     EQUIVL=EQUIVALENT LENGTH
C***  PARM(.,5)=DELX
C***  PARM(.,6)=CHARACTERISTIC IMPEDANCE
C***  PARM(.,7)=VELOCITY OF SOUND IN THE LINE
C     PARM(.,8)=LAMINAR FLOW CONSTANT FOR GIVEN DIA,VISC + DENSITY
C     PARM(.,9)=TURBULENT FLOW CONSTANT FOR GIVEN DIA,VISC + DENSITY
C     PARM(.,2) IS INPUTED AS LINE OD AND CONVERTED TO LINE ID
      REN=1200
      N=1
      DO 80 N=1,NLINE
      IT=LDN(N,7)
      M=N*2
      LC(M)=1
      LC(M-1)=1
      IF(LDN(N,1).LT.10) GO TO 65
      LC(M)=-1
      LDN(N,1)=LDN(N,1)-10
   65 IF(LDN(N,1).EQ.1) GO TO 70
      PARM(N,2)=PARM(N,2)-PARM(N,3)*2.0
      A=1.8*(PARM(N,3)+PARM(N,2)/2.0)*BULK(IT)
      A=BULK(IT)/(RHO(IT)*(1.0+A/(PARM(N,4)*PARM(N,3))))
      IF(LDN(N,1).EQ.2)A=BULK(IT)/RHO(IT)
      GO TO 75
   70 BULKH=BULK(IT)*PARM(N,4)/(BULK(IT)+PARM(N,4))
      A=BULKH/RHO(IT)
   75 A=SQRT(A)
      PARM(N,6)=RHO(IT)*A*4.0/(PI*PARM(N,2)**2)
      CONST=PARM(N,1)/DELT
      IF(CONST.GE.A) GO TO 78
      PC=100.-CONST*100./A
      WRITE(6,300)N,PC
      A=CONST
   78 PARM(N,7)=A
      NLPT(N)=CONST/A+1.01
      PARM(N,5)=PARM(N,1)/(NLPT(N)-1)
      IF(LDN(N,1).EQ.2)GO TO 79
      EQUIVL=PARM(N,2)*(LDN(N,3)*12+LDN(N,4)*57.0+LDN(N,5)/45.*4.65+
     1      (LDN(N,6)/90.+1.)*3.75)+PARM(N,1)*LDN(N,2)/100
      DIA=PARM(N,2)
      PARM(N,8)=128./PI*VISC(IT)*RHO(IT)/(DIA**4.)*(PARM(N,1)+EQUIVL)
      PARM(N,9)=.213*RHO(IT)*(VISC(IT)**.25)/(DIA**4.75)*(PARM(N,1)
     1+EQUIVL)
      GO TO 80
   79 PARM(N,8)=1.0E-6
      PARM(N,9)=1.0E-6
   80 CONTINUE
```

5.3 SUBROUTINE DFRICD

Function DFRICD is used by LINE to calculate pressure loss due to dynamic friction caused by local fluid acceleration. The theory used (see Reference paragraph 9.4) provides an efficient method for determining dynamic friction that gives close correlation with experimental results.

5.3.1 Math Model

The theory used for the basis of this subroutine is described in the paper noted in Reference paragraph 9.4. Equation (29) of the referenced paper is the expression used to describe the combined steady state and dynamic pressure losses under laminar flow conditions.

$$f(t+\Delta t) = \frac{8\rho\nu}{a^2} V(t) + \frac{4\rho\nu}{a^2} (y_1 + y_2 + y_3)$$

Substituting equation (35) into equation (33) produces the following expressions for y_1, y_2 and y_3

$$y_1 (t + \Delta t) = y_1 (t) * e^{-8000*\Delta*\nu/a^2} + 40.* (V (t + \Delta t) - V (t))$$

$$y_2 (t + \Delta t) = y_2 (t) * e^{-200* \Delta t*\nu/a^2} + 8.1* (V (t + \Delta t) - V (t))$$

$$y_3 (t + \Delta t) = y_3 (t)*e^{-26.4*\Delta t*\nu/a^2} + (V (t + \Delta t) - V (t)).$$

Each expression for y at time t + Δt contains a decayed pressure at t added to a pressure loss caused by a change in fluid velocity. Converting velocity to volumetric flow rate and I.R. to I.D. yields

$$f(t+\Delta t) = \frac{128 *\rho*\nu*Q\ (t)}{\pi * D^4} + \frac{16 *\rho*\nu}{D^2} *$$

$$\left[[y_1\ (t)*e^{-8000 * \Delta t*\nu*4/D^2} + \frac{4*40}{\pi*D^2} * (Q\ (t+\Delta t) - Q\ (t))\ ] \right.$$

$$+ [y_2\ (t)*e^{-200 *\Delta t * \nu*4/D^2} + \frac{4*8.1}{\pi*D^2} (Q\ (t + \Delta t) - Q\ (t))\ ]$$

$$\left. + [y_3\ (t)*e^{-26.4 *\Delta t *\nu*4/D^2} + \frac{4}{\Pi*D^2} (Q\ (t + \Delta t) - Q\ (t))\ ] \right]$$

Since the first term in the equation is steady state pressure drop for laminar flow, it is omitted in this subroutine. The dynamic pressure drop equation can then be written

$$\begin{aligned} f(t+\Delta t) = {} & \frac{16*\rho*\nu}{D^2} *y_1\ (t)*e^{-8000*\Delta t*\nu*4/D^2} \\ & + \frac{16 * 160}{\pi*D^4} * (Q\ (t+\Delta t) - Q(t)) \\ & + \frac{16*\rho*\nu}{D^2} * y_2\ (t)*e^{-200*\Delta t*\nu*4/D^2} \\ & + \frac{16 * 32.4}{\pi*D^4} * (Q\ (t+\Delta t) - Q\ (t)) \\ & + \frac{16*\rho*\nu}{D^2} * y_3\ (t)*e^{-26.4*\Delta t*\nu*4/D^2} \\ & + \frac{16*4}{\pi*D^4} * (Q\ (t+\Delta t) - Q\ (t)) \end{aligned}$$

This equation is used to calculate turbulent as well as laminar flow pressure drops under dynamic conditions since little is known about the effects of dynamic friction under turbulent flow conditions.

In the subroutine, pressure loss at time T to be decayed is expressed by:

$$\frac{16*\rho*\nu}{D^2} * Y_{1,2,3} \text{ , and}$$

is actually the latest Y value calculated by entry DFRICU. The first time through the 3000 section of line (i.e., T = DELT), the decayed pressure values are set to zero.

5.3.2 Assumptions

See Reference (4).

5.3.3 Computation Methods

Entry DYFRI is called from the 2000 section of LINE. The purpose of this entry is to calculate and store the dynamic pressure loss equation constants for all lines. The constants are summed and return to the line subroutine, where the value is stored for later use. DFRICD is next called from the 3000 section of LINE. New Y values are calculated using past Y values multiplied by their decay factors. Y1, Y2 and Y3 are summed and set equal to YTOTAL. This decayed value of pressure loss, YTOTAL, at program time T - DELT is returned to LINE which uses it in the first iteration of each time step to calculate flows and pressures at a given interior line point. DFRICU entry is made from the 3000 section of LINE and is used to update the Y values of DFRICD using changes in flow. YTOTAL is set equal to YOLD and all Y values are updated. Y1, Y2 and Y3 are next summed and set to YTOTAL. YOLD is subtracted from YTOTAL and the resulting value is returned to LINE.

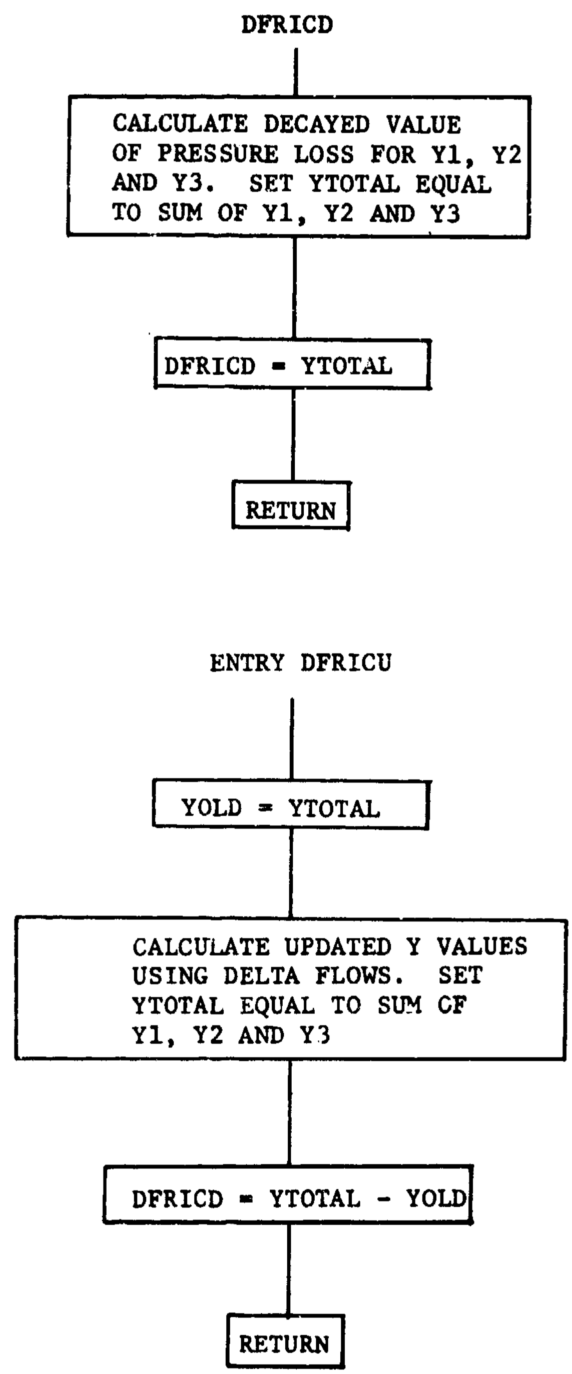
Whenever the LINE segment is cavitated the appropriate decay factors and DFRICD are zeroed.

5.3.4 Approximations

The method of calculating frequency dependent friction is in itself an approximation of the exact expression. The method was chosen to conserve computer core and computation time.

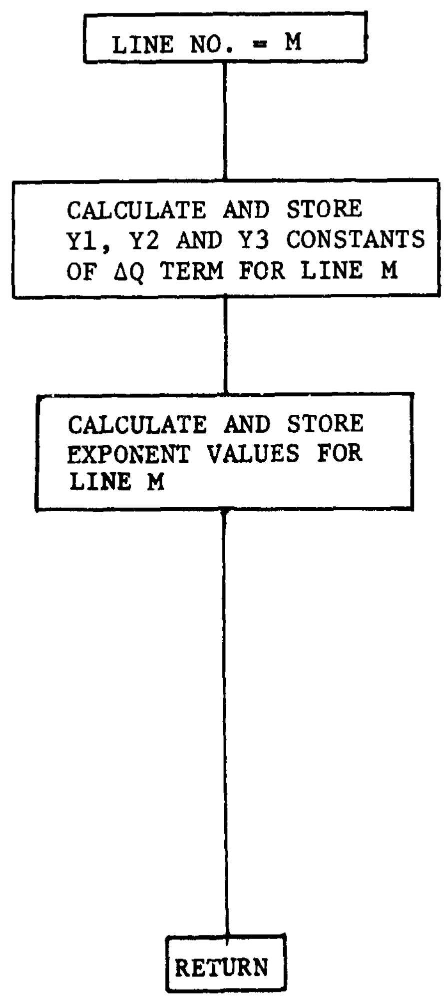
5.3.5 Limitations

See Reference (4).



DFRICD FLOW DIAGRAM
FIGURE 5.3-1

ENTRY DYFRI



ENTRY DYFRI FLOW DIAGRAM
FIGURE 5.3-2

5.3.6 Variable Names

Variable	Description	Units
CONST	Constant in dynamic pressure loss equation	--
DELQ	New flow - Old flow	CIS
DFR1CD	Dummy variable	--
DIA	I.D. of tube	IN
EX1	Constant in dynamic pressure loss equation	--
EX2	Constant in dynamic pressure loss equation	--
EX3	Constant in dynamic pressure loss equation	--
JJ	Address LSTART(N) + NLPT(N) -1	--
M	Address LSTART(N)	--
MM	Address LSTART(N)	--
MPT	Address LSTART(N) +1	--
M	Counter indicating assigned line number	--
YOLD	Dummy variable	--
YTOTAL	Decayed value of pressure	PSI
Y1 ( )	Array for Y1 valves	--
Y2 ( )	Array for Y2 values	--
Y3 ( )	Array for Y3 values	--

5.3.7 Subroutine Listing

```
      FUNCTION DFRICD(DELQ,MPT,JJ,M)
C *** REVISED AUGUST 5, 1975 ***
      COMMON/SUB/PARM(150,9),PM(1500),QM(1500),P(300),Q(300),C(300)
     1,Z(300),RHO(20),S2ORHO(20),VISC(20),BULK(20),TEMP(20),PVAP(20)
     2,ATPRES,T,DELT,TFINAL,PLTDEL,PI,TITLE(20),LEGN,ICON
     3,KTEMP(99),LSTART(150),NLPT(150),LTYPE(99),NC(99),INX,INZ
     4,INV,ISTEP,NLINE,NEL,IND,IENTR,MNLINE,MNEL,MNLEG,MNNODE,MNPLOT
     5,MNLPTS,MDS
      DIMENSION Y1(1500),Y2(1500),Y3(1500)
      DATA Y1,Y2,Y3/4500*0.0/
      IF(M.LT.0)GO TO 10
      Y1(MPT)=Y1(MPT)*Y1(JJ)
      Y2(MPT)=Y2(MPT)*Y2(JJ)
      Y3(MPT)=Y3(MPT)*Y3(JJ)
      YTOTAL=Y1(MPT)+Y2(MPT)+Y3(MPT)
      DFRICD=YTOTAL
C     DFRICD=0.0
      RETURN
   10 Y1(MPT)=0.0
      Y2(MPT)=0.0
      Y3(MPT)=0.0
   20 DFRICD=0.0
      RETURN
      ENTRY DFRICU
      IF(JJ.LT.0)GO TO 20
      YOLD=YTOTAL
      Y1(MPT)=Y1(MPT)+DELQ*Y1(M)
      Y2(MPT)=Y2(MPT)+DELQ*Y2(M)
      Y3(MPT)=Y3(MPT)+DELQ*Y3(M)
      YTOTAL=Y1(MPT)+Y2(MPT)+Y3(MPT)
      DFRICD=YTOTAL-YOLD
C     DFRICD=0.0
      RETURN
      ENTRY DYFRI
C     IBM 360 - ZERO Y1,Y2,Y3 ARRAYS HERE
      RAD=PARM(M,2)/2.0
      MM=LSTART(M)
      JJ=MM+NLPT(M)-1
      CONST=PARM(M,7)*DELT*RHO(MPT)*VISC(MPT)/(PI*RAD**4.)
      Y1(MM)=160.*CONST
      Y2(MM)=32.4*CONST
      Y3(MM)=4.*CONST
      CONST=DELT*VISC(MPT)/RAD**2.
      EX1=-8000.*CONST
      EX2=-200.*CONST
      EX3=-26.4*CONST
      Y1(JJ)=EXP(EX1)
      Y2(JJ)=EXP(EX2)
      Y3(JJ)=EXP(EX3)
      DFRICD=Y1(MM)+Y2(MM)+Y3(MM)
      RETURN
      END
```

APPENDIX R
TYPE #61 CONSTANT PRESSURE RESERVOIR
HYTTHA USER MANUAL (AFAPL-TR-76-43, VOL. VII)

6.61 TYPE #61 CONSTANT PRESSURE RESERVOIR

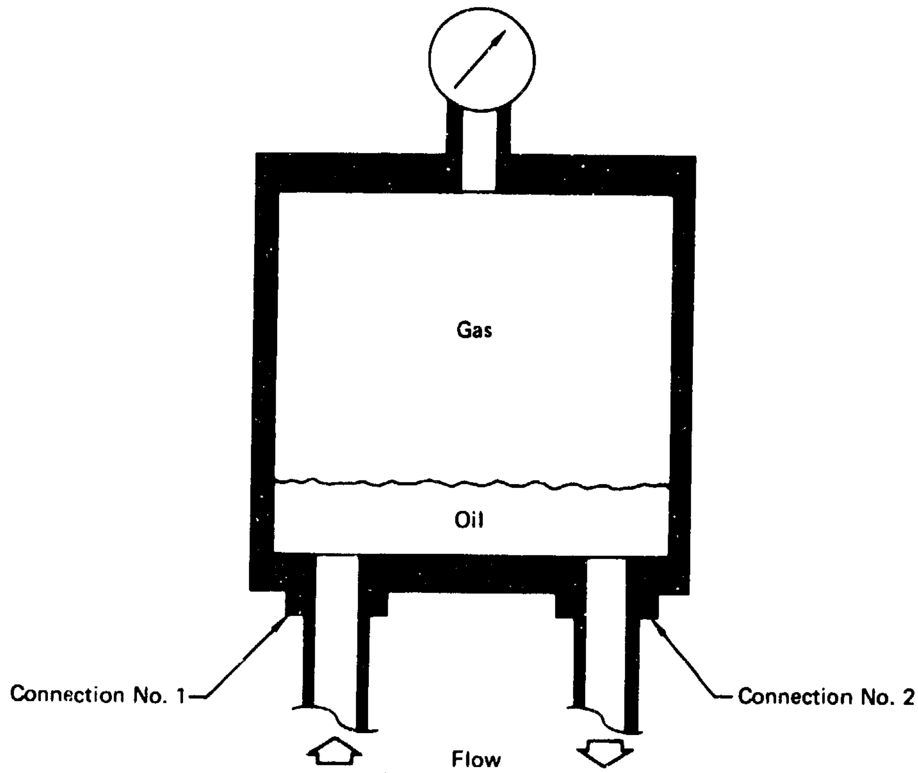


FIGURE 6.61-1
TYPE NO. 61 CONSTANT PRESSURE RESERVOIR

The Type #61 constant pressure reservoir which is used for test simulation purposes, requires only the connection information the reservoir pressure, fluid and wall temperatures. Any of the four connections not used are blanked off. The temperatures are considered to be constant.

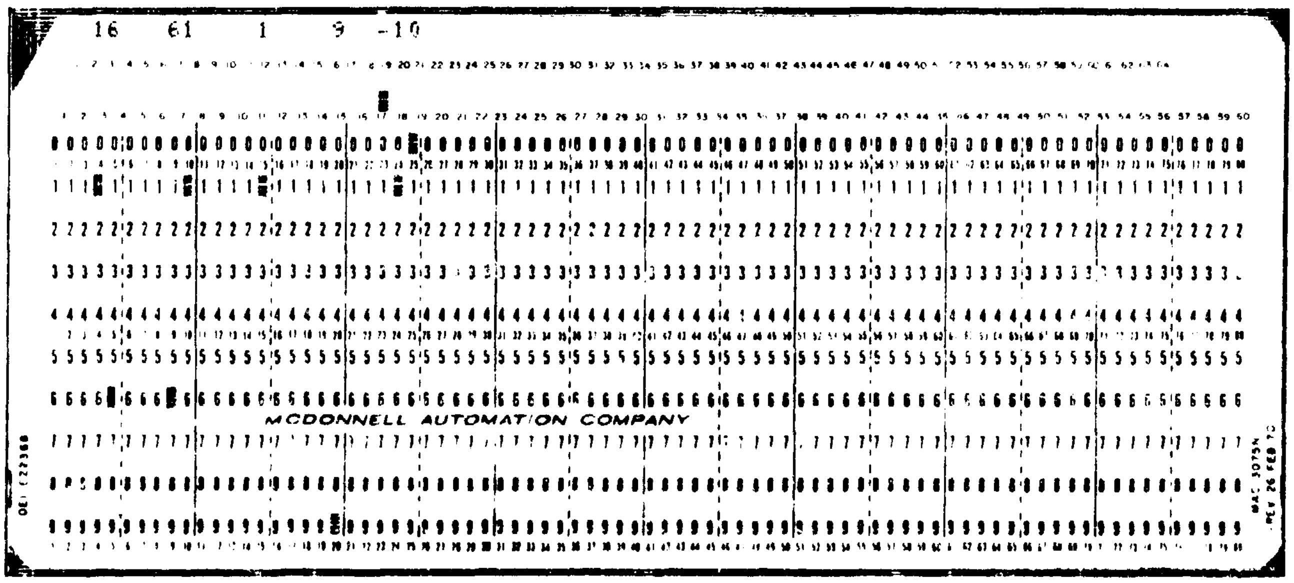
The user may input a variable fluid temperature profile by specifying the number of data points in columns 56-60 of card number one. The fluid temperature at a particular time will be derived from the tabulated input data input on the third and fourth data cards.

TRSVR61

CARD NUMBER 1

COLUMN	FORMAT	DATA
1-5	I5	Component Number
6-10	I5	Type Number = 61
11-15	I5	Number of Real Data Cards = 1
16-20	I5	Line Number (with sign) attached to Connection 1
21-25	I5	Line Number (with sign) attached to Connection 2
26-30	I5	Line Number (with sign) attached to Connection 3
31-35	I5	Line Number (with sign) attached to Connection 4
36-40	I5	Default Temperature Indicator
41-45	I5	Environment Number
46-50	I5	Number of Temperature Data Points
51-55	I5	
56-60	I5	
61-65	I5	
66-70	I5	
71-75	I5	
76-80	I5	

EXAMPLE CARD

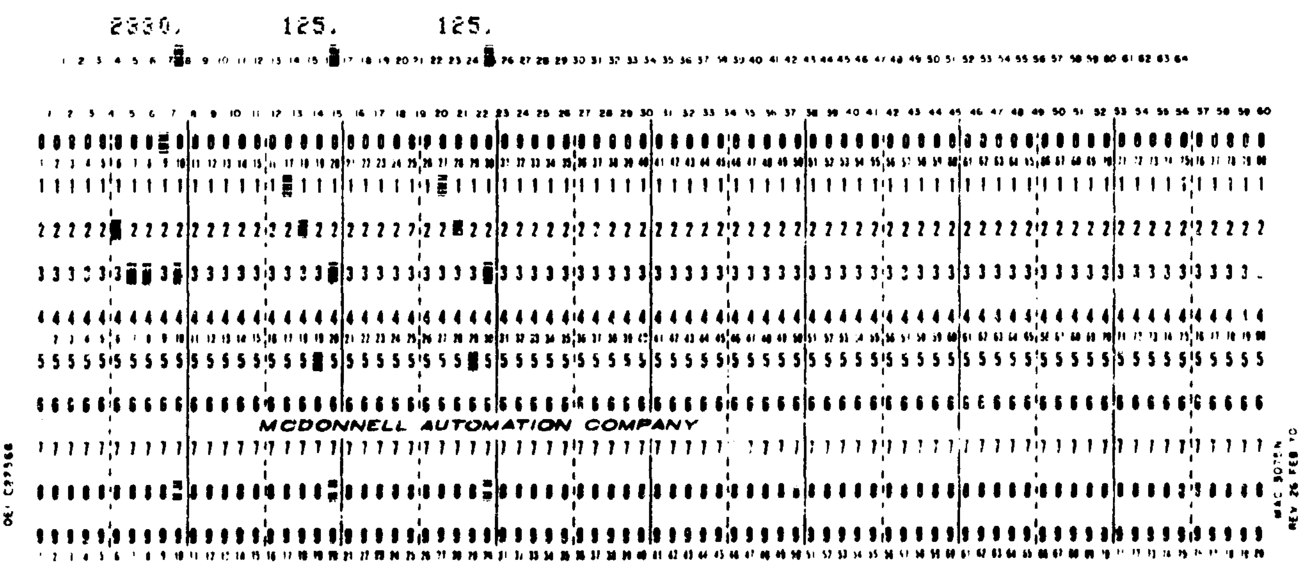


TRSVR61

CARD NUMBER 2

COLUMN	FORMAT	DATA	DIMENSIONS
1-10	E10.0	Reservoir Pressure	PSI
11-20	E10.0	Fluid Temperature	°F
21-30	E10.0	Wall Temperature	°F
31-40	E10.0		
41-50	E10.0		
51-60	E10.0		
61-70	E10.0		
71-80	E10.0		

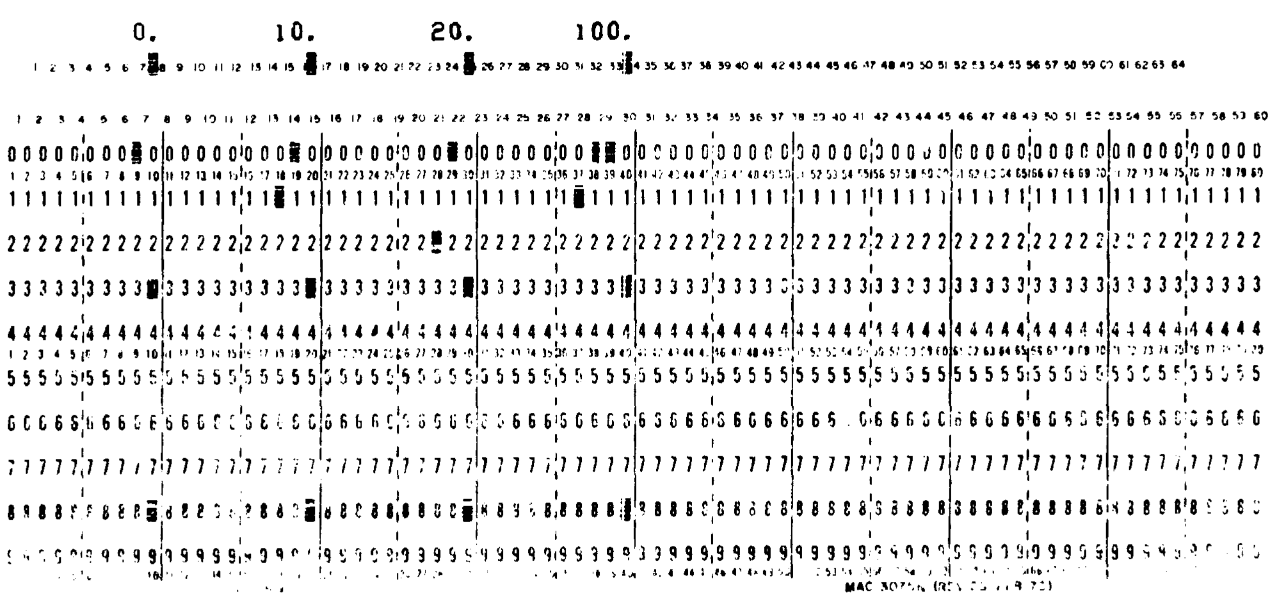
EXAMPLE CARD



CARD NUMBER

COLUMN	FORMAT	DATA	DIMENSIONS
1-10	E10.0	First Time Value - Should be 0.0	SEC
11-20	E10.0	(Enter as Many Time Values as Required	
21-30	E10.0	Using as Many Columns and Cards as Necessary	
31-40	E10.0	- Final Time Should be Greater than or	
41-50	E10.0	Equal to the Final Calculation Time).	
51-60	E10.0		
61-70	E10.0		
71-80	E10.0		

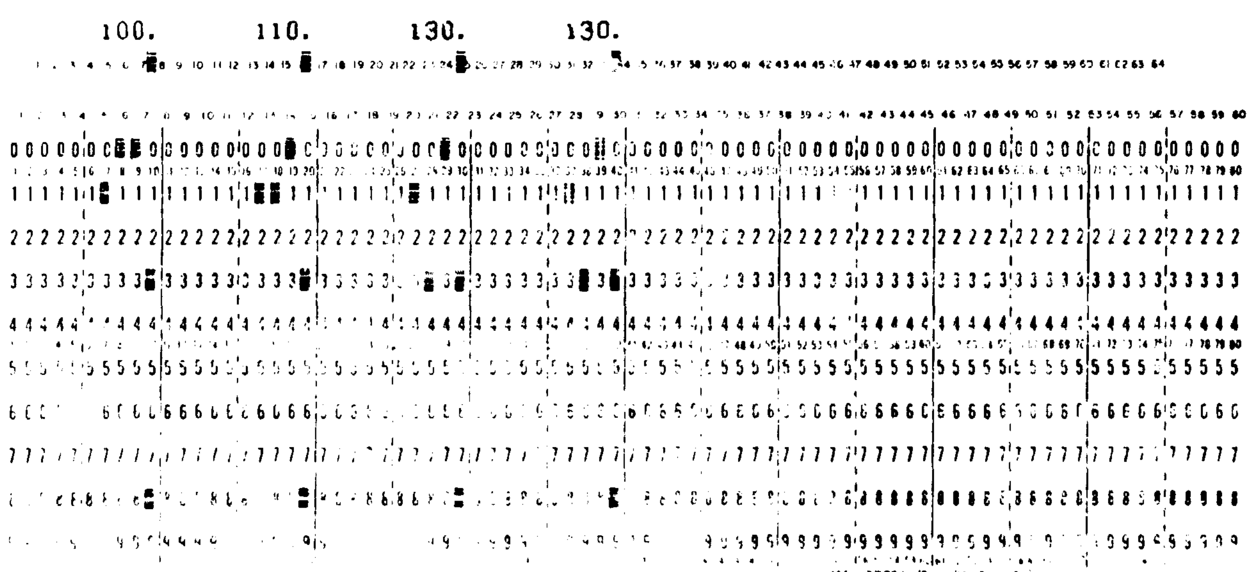
EXAMPLE CARD



CARD NUMBER

COLUMN	FORMAT	DATA	DIMENSIONS
1-10	E10.0	Initial Temperature @ T = 0.0	DEG F
11-20	E10.0	(Enter as Many Temperatures as Time	
21-30	E10.0	Values).	
31-40	E10.0		
41-50	E10.0		
51-60	E10.0		
61-70	E10.0		
71-8C	E10.0		

EXAMPLE CARD



6.61 SUBROUTINE TRSVR61

TRSV61 models a hypothetical constant pressure, constant temperature reservoir that can be used in test simulation work as sketched in Figure 6.61-1. The input pressure is maintained without fluctuation while the flow rates are adjusted to meet the line requirements. A maximum of four connections can be used.

This subroutine calculates the fluid and wall temperatures of the component at each connection.

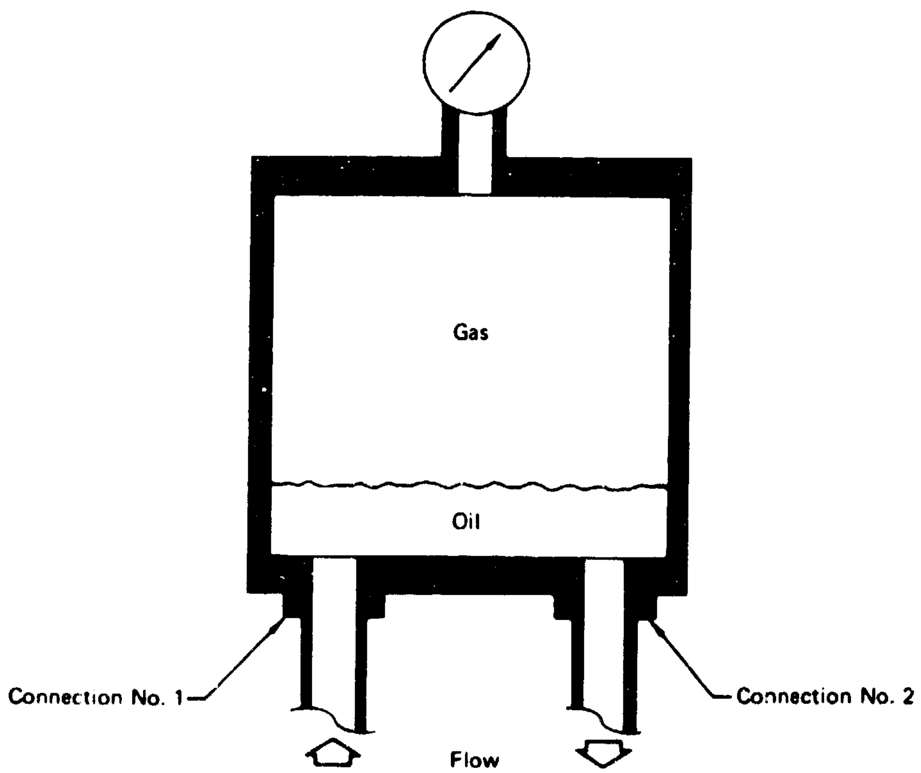


FIGURE 6.61-1
TYPE NO. 61 CONSTANT PRESSURE RESERVOIR

6.61.1 Math Model

The thermal math model for the reservoir includes heat transfer to and from one to four connecting line segments. They can be either downstream or upstream of the reservoir. To understand TRSV61 we shall look at a hydraulic system with two reservoirs connected by a line as shown in Figure 6.61-2. The line is downstream of reservoir one and

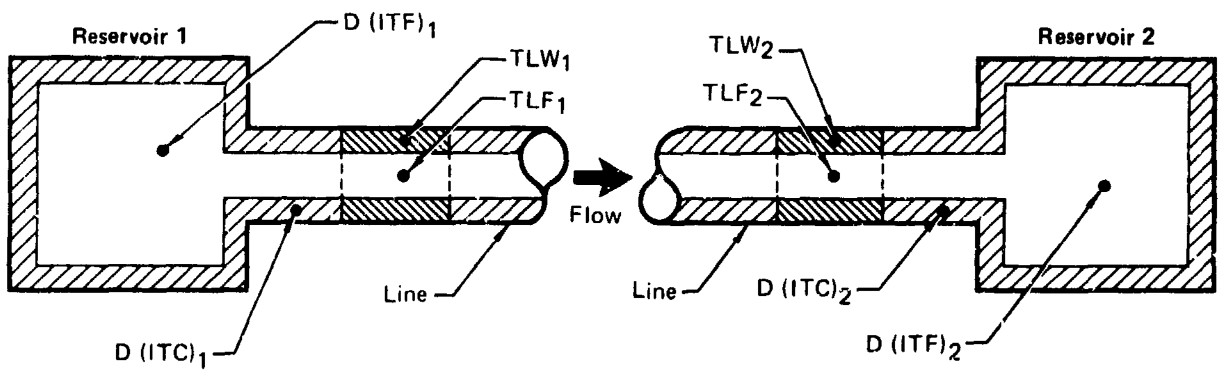


FIGURE 6.61-2
RESERVOIR NODE REPRESENTATION FOR SAMPLE SYSTEM

3P77-0065-4

upstream of reservoir two. As is discussed in subroutine TLINEA, the line is divided into equal segments. In Figure 6.61-2 the temperatures of the fluid and wall of reservoir 1 are D(ITF), and D(ITC), the temperature of the fluid and wall of the line segment connecting reservoir 1 are TLF_1 and TLW_1, the temperatures of the fluid and wall of the line segment connecting reservoir 2 are TLF2 and TLW_2, and the temperatures of the fluid and wall of

reservoir 2 are $D(ITF)_2$ and $D(ITC)_2$. For downstream connecting line segments, such as these similar to segment 1 in Figure 6.61-2, the subroutine assigns the temperatures of the reservoir fluid and wall as end conditions of the reservoir, and boundary conditions of the first line segment,

$$TF(L1) = D(ITF)_1$$

$$TW(L1) = D(ITC)1$$

For upstream connecting line segments, such as those similar to segment 2 in Figure 6.61-2, the subroutine assigns the temperatures of the reservoir wall to the reservoir connection, or the boundary condition of the line segment

$$TW(L2) = D(ITC)_2$$

and also assigns the temperatures of the fluid entering the reservoir to the temperature of the line segment, $TF(L2) = TLF_2$. Note however that the temperature of the fluid in reservoir 2 is $D(ITF)_2$ and eventually the fluid entering reservoir 2 will equal TLF_2. In the hydraulic math model the input constant reservoir pressure is assigned to the reservoir node number.

6.61.2 Assumptions

1. Fluid and wall temperatures of the reservoir remain constant, except the fluid entering a reservoir makes the reservoir fluid the same temperature as the fluid in the connecting line.
2. The reservoir is assumed to have an infinitely large gas volume so that the pressure remains unchanged.
3. The interface conductance between the reservoir walls and the line walls is infinite.

6.61.3 Computation Methods

SECTION 1000

The number of active reservoir connections is determined from the NC( ) array. A DO loop is then set up to initialize all the connecting line wall and fluid temperatures.

SECTION 2000

The node number of the reservoir is determined and the flow into and/or out of the reservoir is summed for each active connection.

D(4) = D(4) + QR

Counter, L(6), is incremented by 1 each time an entry is made until the counter is equal to the number of active connections. Once the total net flow has been determined, QN(N) is calculated

QN(N) = D(PRESS) * 10000.-D(4)

An external pressure array is set to a constant value

PEX(N) = 10000.

The total flow and counter, L(6), are then set to zero.

SECTION 3000

The number of connecting line segments and flow direction are first determined. Property values are assigned. The exiting fluid and all wall connection temperatures are assigned. The assigned values are put into arrays TC and TF in /TRANS/.

If the user has specified a variable fluid temperature, a call is made to the INTERP subroutine.

IF(L(9),GT,0) CALL INTERP(TIME,D(9),D(L(10)),10,L(9),TF(N),IERR)

INTERP will perform a linear interpolation between the data points and return the fluid temperature, TF(N) values. If the flow is entering the reservoir, INTERP will not be called.

6.61.4 Variable Listing

Variable	Description	Dimension
D(ITC)	Initial temperature of the reservoir walls	°F
D(ITF)	Initial temperature of the reservoir fluid	°F
Q(LI)	Flow in connector I	CFS
N	Reservoir node number	--
D(PRESS)	Pressure of reservoir	PSI
TF(LI)	Temperature of fluid leaving reservoir	°F
TC(LI)	Temperature of reservoir wall connected to connection I	°F

```
      SUBRO[illegible]NE TR[illegible] [illegible]D,DT,DD[illegible]
C *** REVISED SEPTEMBER 17, 1979 ***
      COMMON/TRANS [illegible],TF[illegible]
     + [illegible],DXF(300),TIME,DELT,PI,NLINE,NEL
      COMMON [illegible]TYPE(99),N[illegible](99),KTEMP(99),IND,IENTA,INEL
      COMMON [illegible]TEA[illegible]N(90),QN(90),PE[illegible](90),POLEG(90)[illegible](90)[illegible]
     + [illegible]A,[illegible]N,NNODE,NLEG,NCPR,TERM,
     + LEGN,I[illegible]N,INV,INX,INZ,NUP(90),NDWN(90),NELEM([illegible]
     [illegible]
      COMMON/FLUID/ATPRES,CP,CPFN,FTEMP,PROP(13,3),IDAT,AT,IDSTT,STT,
     +IDWT,WT,IDTEMP,IDFLT,FLT
      COMMON/ENVIRON/NENV,VAT(5),VSTT(5),VWT(5)
      INTEGER PRESS
      DIMENSION D(10),DT(1),DD(1),L(8)
      DATA PRESS/1/,ITF/2/,ITC/3/
      IF(IEN[illegible]) 1000,2000,3000
 1000 CONTINUE
      IF(IDTEMP .EQ. 0 .OR. L(5) .EQ. 0) GOTO 1100
      IF(IDFLT .EQ. 1) D(2)=FLT
      IF(IDWT .EQ. 1) D(3)=WT
 1100 IF(L(6).EQ.0)GOTO 333
C *** SET DATA TO PROPER ENVIRONMENT ****
      D(ITC)=VWT(L(6))
C
  333 D(4)=0.0
      NCI=NC(IND)
      L(5)=NCI
      L(6)=0
C     L(7)= NUMBER OF TIME VALUES
      LL=(L(7)+7)/8
      L(8)=9+LL*8
      DO 1010 I=1,NCI
      N=L(I)
      TF(N)=D(ITF)
      TC(N)=D(ITC)
 1010 CONTINUE
      RETURN
```

```
 2000 CONTINUE
      N=NDWN(INEL)
      QR=Q1
C     INX = ELEMENT NUMBER IN LEG
      IF(INX.NE.1) GO TO 1600
      QR=-QR
      N=NUP(INEL)
 1600 D(4)=D(4)+QR
      L(6)=L(6)+1
C     WRITE(6,900)INEL,N,Q1,PUP
  900 FORMAT(10X,*RSVR*,2I10,2E12.5)
      IF(L(6).NE.L(5))RETURN
      QN(N)-D(PRESS)*10000.-D(4)
      PEX(N)=10000.0
      D(4)=0.0
      L(6)=0
C     PN(N)=D(PRESS)
      RETURN
 3000 CONTINUE
      NCI=L(5)
      DO 3009 I=1,NCI
      N=L(I)
      IF(Q(N).GT.0.0) GO TO 3006
      TF(N)=D(ITF)
 3006 TC(N)=D(ITC)
      IF(L(7).GT.0)CALL INTERP(TIME,D(9),D(L(8)),10,
     + L(7),TF(N),IERR)
 3009 CONTINUE
      RETURN
      END
```

6.51 TYPE #51 - PUMP

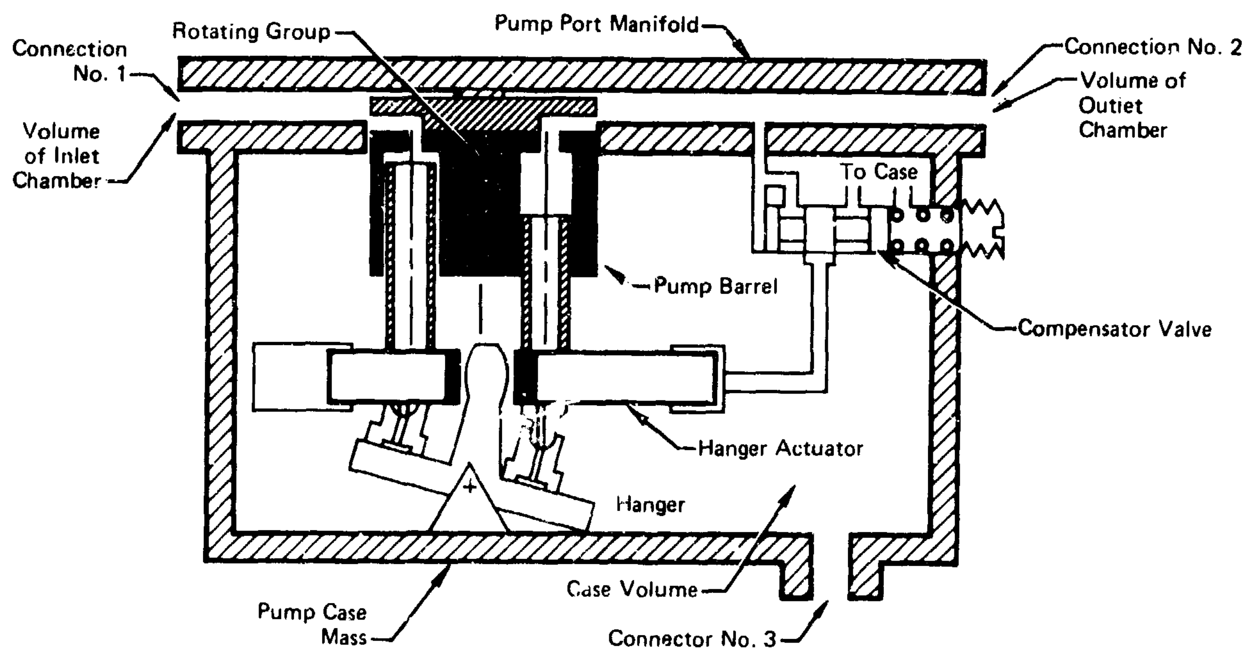


FIGURE 6.51-1
TYPE NO. 51 PRESSURE REGULATED VARIABLE DISPLACEMENT PUMP

GP77-0065-19

The pump subroutine TPUMP51 is a simulation of a pressure regulated variable displacement pump. The pump model computes values of outlet and case drain flows and pressures.

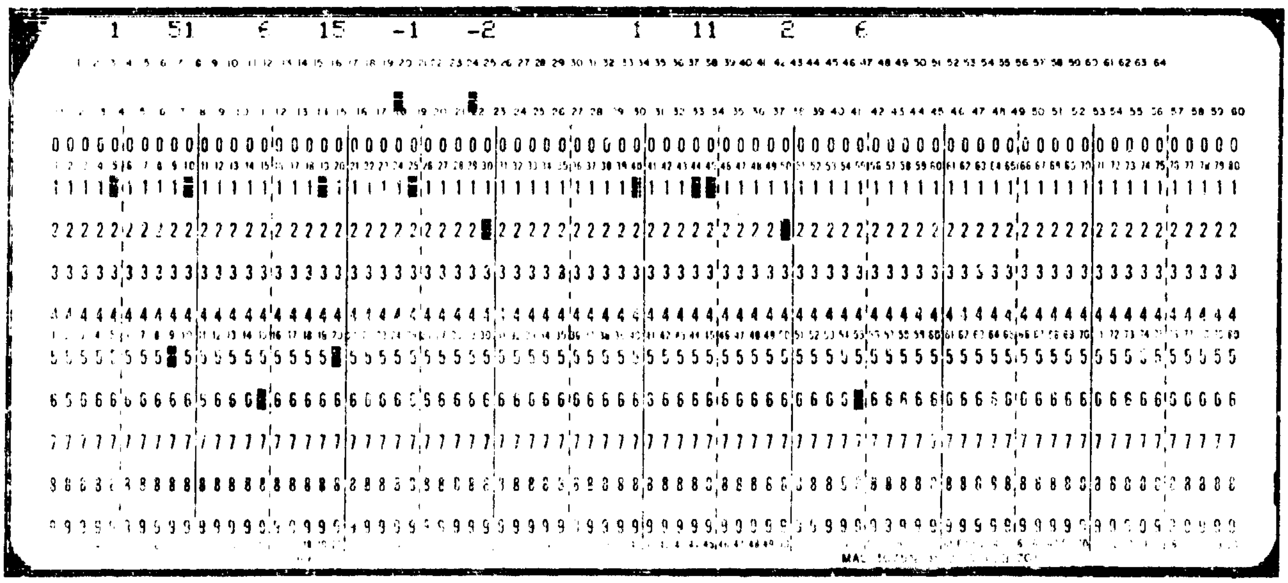
In the thermal model large masses have been grouped together to simplify the simulation. Three large masses are used, the internal moving parts, the walls of the pump around the case volume, and the pump wall at the port manifold. The calculated variables are the rotating group and case temperatures, and three fluid temperatures; the exit fluid, the case drain fluid, and the inlet fluid.

TPUMP51

CARD NUMBER 1

COLUMN	FORMAT	DATA
1-5	I5	Component Number
6-10	I5	Type Number = 51
11-15	I5	Number of Real Data Cards = 4
16-20	I5	Line Number (with sign) attached to Connection 1
21-25	I5	Line Number (with sign) attached to Connection 2
26-30	I5	Line Number (with sign) attached to Connection 3
31-35	I5	Not used
36-40	I5	Leg Number Containing Connection 1
41-45	I5	Leg Number Containing Connection 2
46-50	I5	Leg Number Containing Connection 3
51-55	I5	Number of Data Points in Table
56-60	I5	Default Temperature Indicator
61-65	I5	Environment Number
66-70	I5	
71-75	I5	
76-80	I5	

EXAMPLE CARD

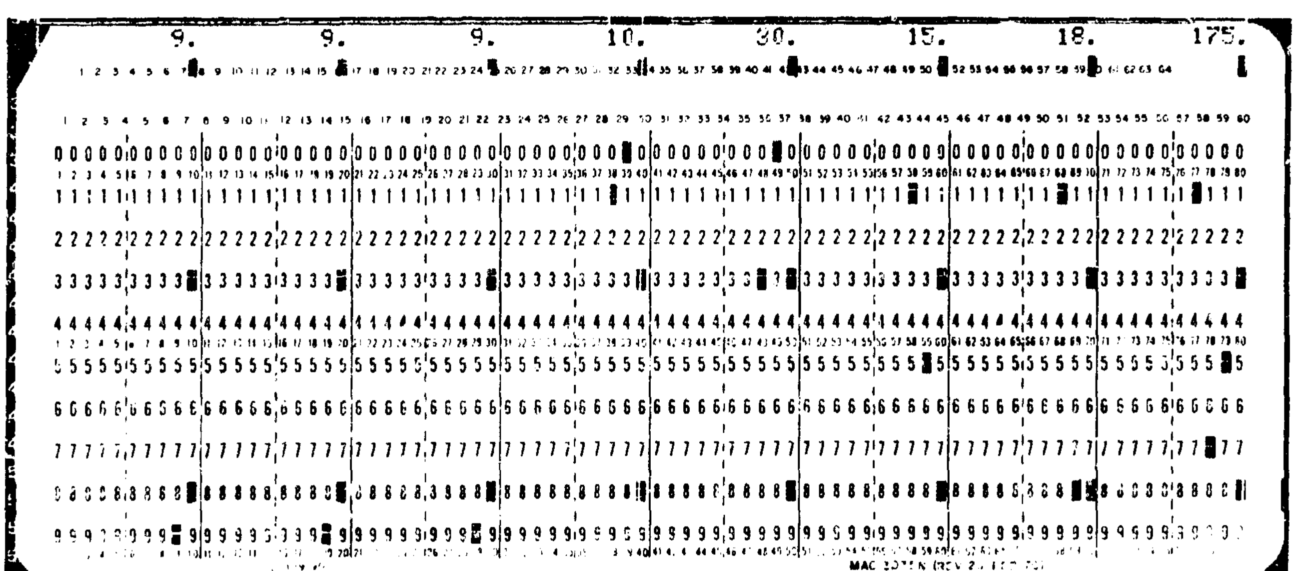


CARD NUMBER 2

COLUMN	FORMAT	DATA	DIMENSIONS
1-10	E10.0	PUMP WALLS MATERIAL TYPE	--
11-20	E10.0	ROTATING GROUP* MATERIAL TYPE	--
21-30	E10.0	PUMP PORT MANIFOLD MASS (SURROUNDING VOL 1&VOL2)	LBm
31-40	E10.0	ROTATING GROUP MASS*	LBm
41-50	E10.0	PUMP CASE MASS	LBm
51-60	E10.0	PUMP INLET VOLUME	$IN.^3$
61-70	E10.0	PUMP OUTLET VOLUME	$IN.^3$
71-80	E10.0	PUMP CASE VOLUME	$IN.^3$

* The rotating group consists of all moving parts, as the pistons, yoke, drive shaft, etc.

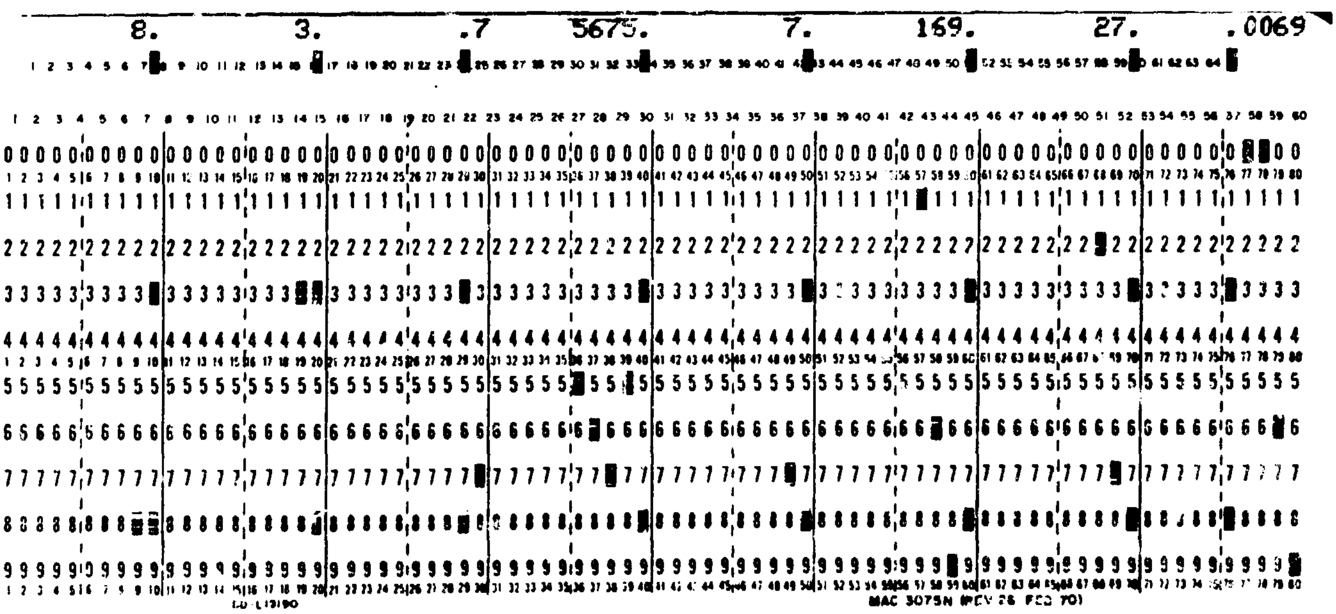
EXAMPLE CARD



CARD NUMBER 3

COLUMN	FORMAT	DATA	DIMENSIONS
1-10	E10.0	AVERAGE CROSS SECTIONAL AREA OF THE ROTATING GROUP	$IN.^2$
11-20	E10.0	CONTACT AREA BETWEEN THE ROTATING GROUP AND THE PUMP WALLS	$IN.^2$
21-30	E10.0	HEAT TRANSFER COEFFICIENT BETWEEN EXIT FLUID AND ROTATING GROUP	$WATTS/IN^2$-°F
31-40	E10.0	HEAT REJECTION AT RATED FLOW	WATTS
41-50	E10.0	DISTANCE FROM INLET TO OUTLET THRU PORT PLATE	IN.
51-60	E10.0	EXTERNAL SURFACE AREA OF PUMP PORT MANIFOLD	$IN.^2$
61-70	E10.0	EXTERNAL SURFACE AREA OF PUMP CASE	$IN.^2$
71-80	E10.0	HEAT TRANSFER, COEFF. FROM ATMOSPHERE TO PUMP CASE	$WATTS/IN^2$-°F

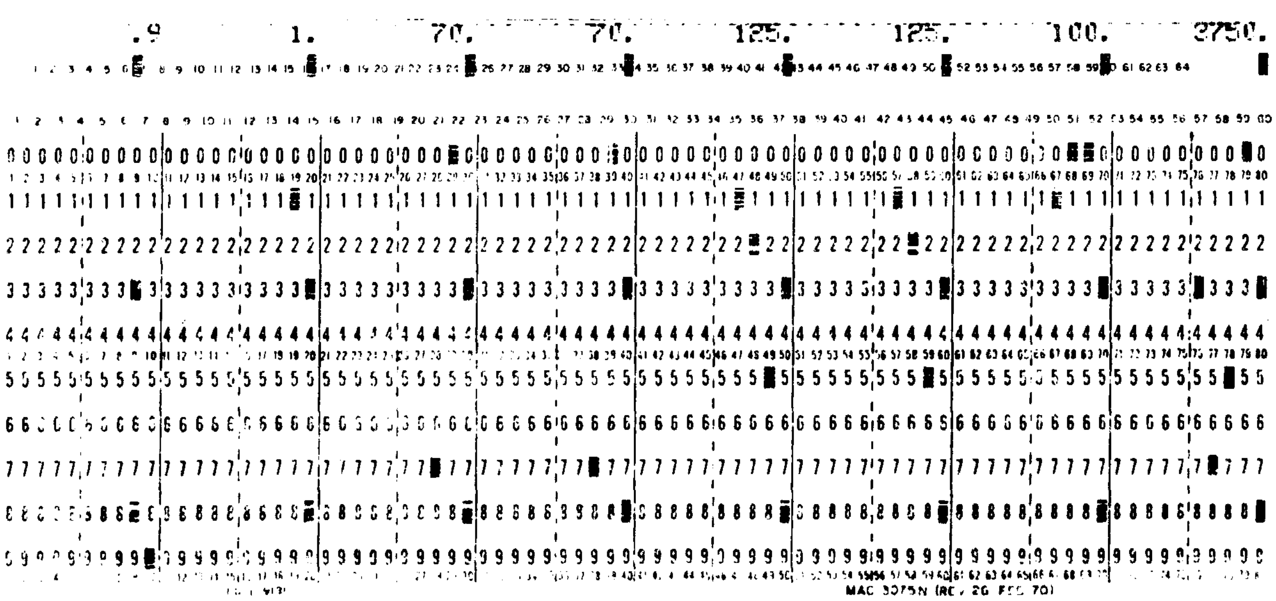
EXAMPLE CARD



CARD NUMBER 4

COLUMN	FORMAT	DATA	DIMENSIONS
1-10	E10.0	INTERFACE CONDUCTANCE BETWEEN THE CASE AND THE ROTATING GROUP	WATTS/IN^2-°F
11-20	E10.0	HEAT TRANSFER COEFF., INLET FLUID TO MANIFOLD WALLS	WATTS/IN^2-°F
21-30	E10.0	SURROUNDING STRUCTURE TEMPERATURE	°F
31-40	E10.0	SURROUNDING ATMOSPHERIC TEMPERATURE	°F
41-50	E10.0	INITIAL TEMPERATURE OF THE FLUID	°F
51-60	E10.0	INITIAL TEMPERATURE OF THE WALLS	°F
61-70	E10.0	RATED FLOW	CIS
71-80	E10.0	PUMP RPM AT RATED FLOW	RPM

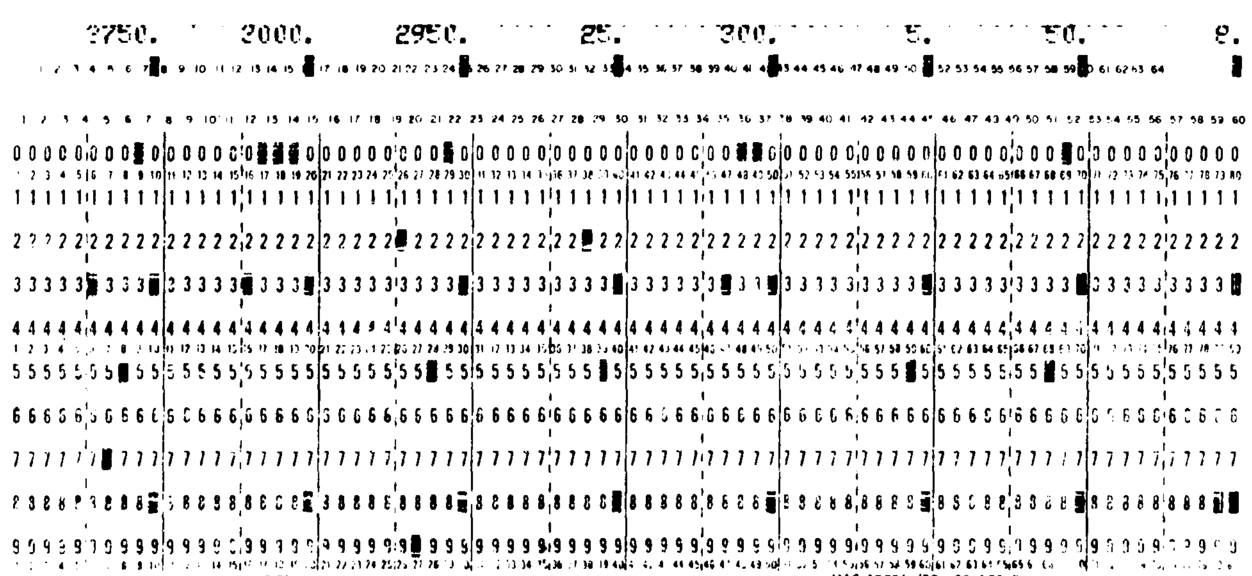
EXAMPLE CARD



CARD NUMBER 5

COLUMN	FORMAT	DATA	DIMENSIONS
1-10	E10.0	PUMP SPEED	RPM
11-20	E10.0	RATED OUTPUT PRESSURE AT ZERO FLOW	PSI
21-30	E10.0	RATED OUTPUT PRESSURE AT FULL FLOW	PSI
31-40	E10.0	MINIMUM INLET PRESSURE	PSI
41-50	E10.0	MAXIMUM PRESSURE DIFFERENCE BETWEEN PUMP CASE AND INLET	PSID
51-60	E10.0	CASE FLOW AT RATED FLOW AND PRESSURE	CIS
61-70	E10.0	CASE PRESSURE AT RATED FLOW AND PRESSURE	PSI
71-80	E10.0	DEPTH OF PUMP CASE	IN

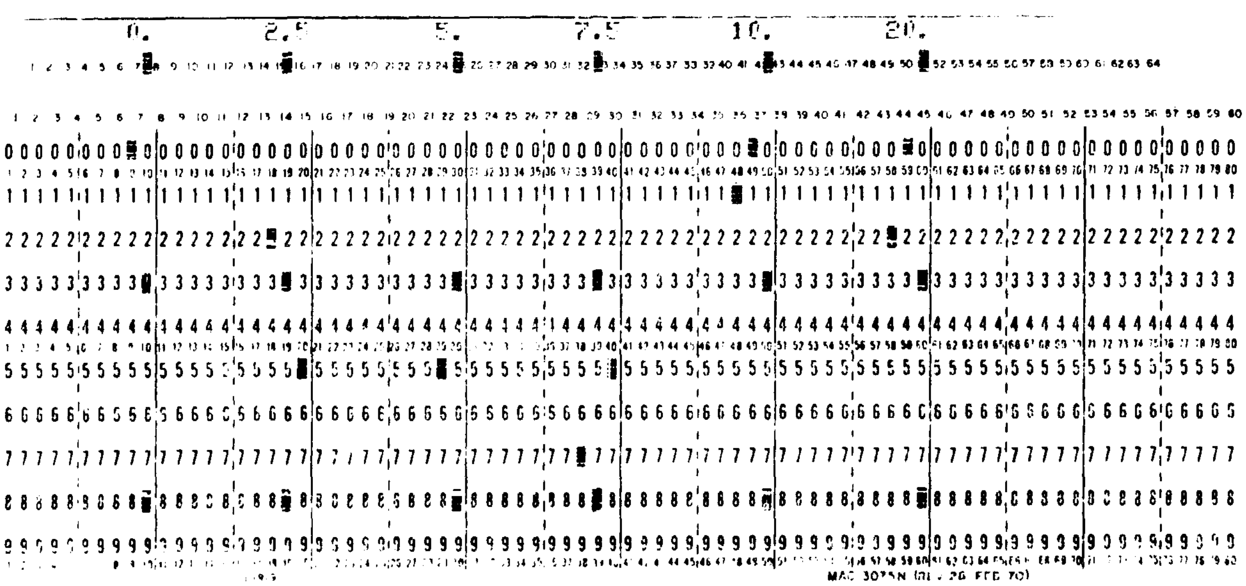
EXAMPLE CARD



CARD NUMBER 6

COLUMN	FORMAT	DATA	DIMENSIONS
1-10	E10.0	First Pump RPM Time - Should be 0.0	SEC
11-20	E10.0	(Enter as Many Time Values As Required	SEC
21-30	E10.0	Using As Many Columns and Cards as Necessary	
31-40	E10.0	- Final Time Should Be the Final Calcula-	
41-50	E10.0	tion Time).	
51-60	E10.0		
61-70	E10.0		
71-80	E10.0		

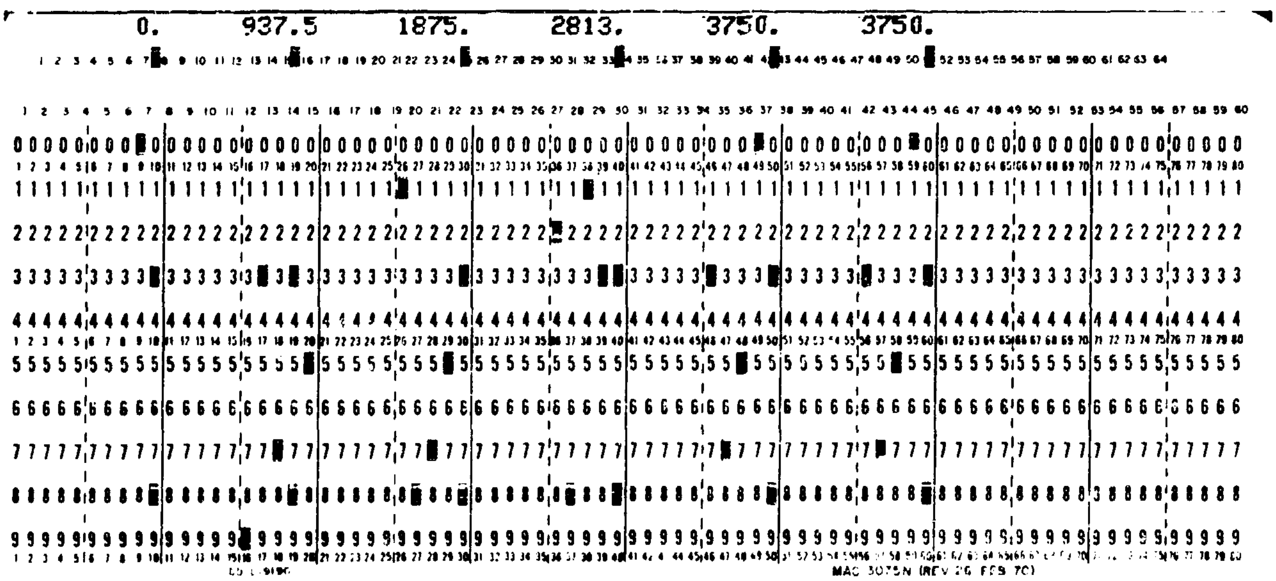
EXAMPLE CARD



CARD NUMBER 7

COLUMN	FORMAT	DATA	DIMENSIONS
1-10	E10.0	Initial Pump RPM @ T = 0.0	REV./MIN.
11-20	E10.0	(Enter as Many Pump RPM Values as Time	
21-30	E10.0	Values).	
31-40	E10.0		
41-50	E10.0		
51-60	E10.0		
61-70	E10.0		
71-80	E10.0		

EXAMPLE CARD



6.51 Subroutine TPUMP51

Subroutine TPUMP51 simulates a variable displacement inline piston pump sketched in Figure 6.51-1. The subroutine calculates the temperatures of the exit fluid, the inlet chamber fluid, the case fluid, the internal moving parts (assumed one node), and the pump walls.

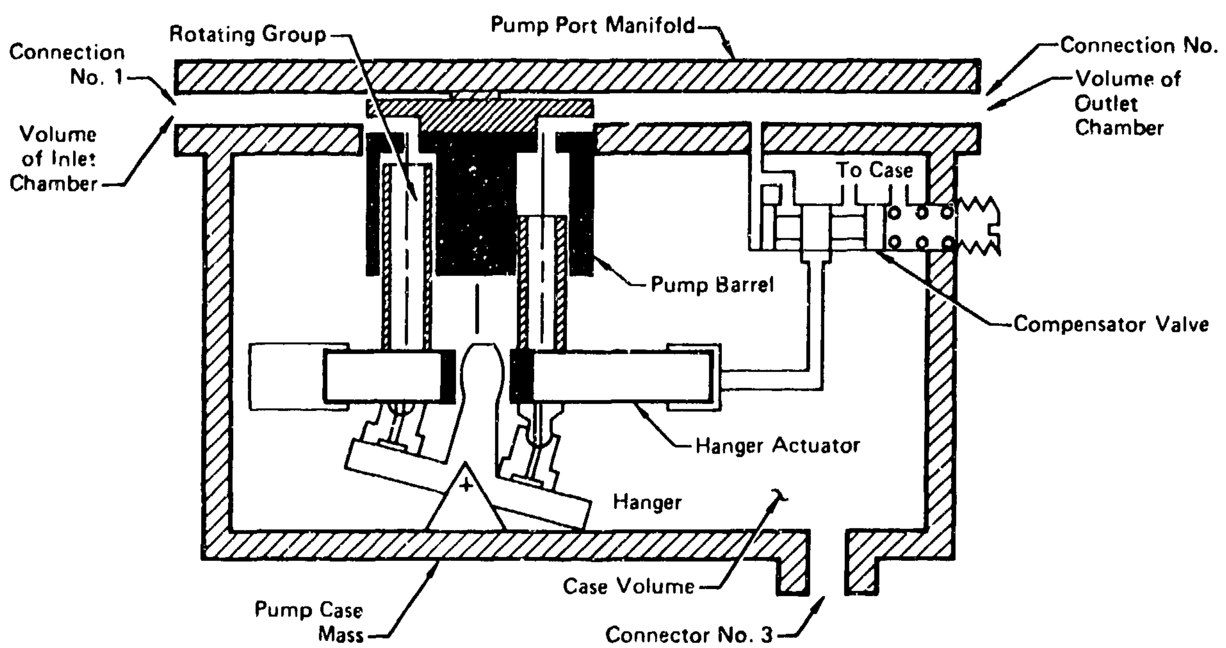


FIGURE 6.51-1
TYPE NO. 51 PRESSURE REGULATED VARIABLE DISPLACEMENT PUMP

GP77-0065-19

6.51.1 Math Model

The thermal math model for the pump includes heat transfer to and from three connecting line segments, one upstream, one downstream and one at the case drain. Thirteen nodes are considered: six fluid nodes, six wall nodes, and one node for the internal moving parts of the pump, called the piston node (as shown in Figure 6.51-2). The pump consists of

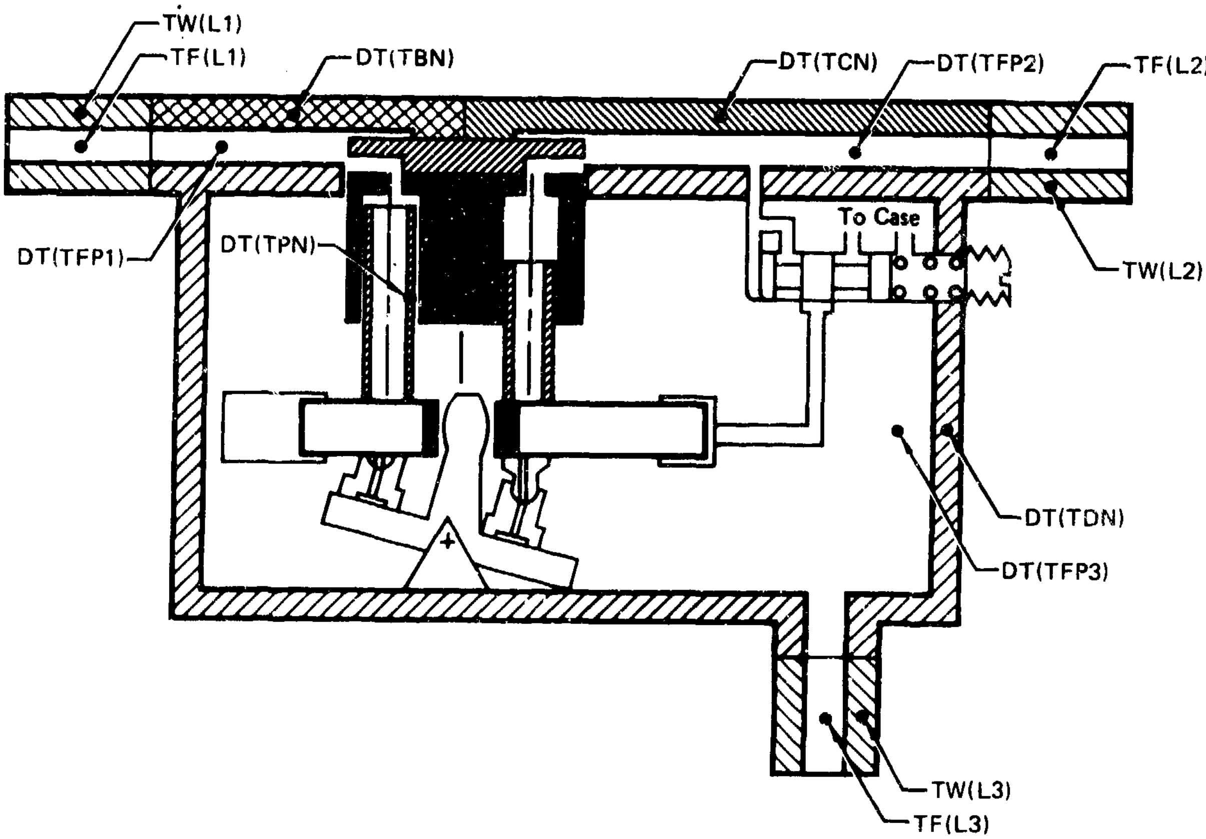


FIGURE 6.51-2
TYPE NO. 51 PRESSURE REGULATED VARIABLE DISPLACEMENT PUMP AND LINE NODE REPRESENTATION

GP77-0065-22

seven nodes: three fluid (one inlet, one outlet, and one case), three walls, one inlet, one outlet, and one around the case drain, and one node for the internal moving parts of the pump, the piston.

The temperatures of the three line segment nodes are TF(L1) and TW(L1), TF(L2) and TW(L2), and TF(L3) and TW(L3) for the inlet segment, exit segment, and case drain line segment fluid and wall nodes respectively. The pump

inlet volume temperature is DT(TFP1), exit volume temperature is DT(TFP2), the case drain fluid volume temperature is by DT(TFP3), the pump wall temperature, around the inlet, is denoted by DT(TBN), the wall temperature around the exit is DT(TCN), the wall temperature around the case drain is DT(TDN), and the pistons temperature denoted by DT(TPN).

Seven heat balance equations are written to solve for the seven pump node temperatures, using the pump and line segment material properties and dimensions, the atmosphere and structure temperatures external to the pump, and TF(L1), TW(L1), TW(L1), TW(L2) and TW(L3). (Note TF(L2) = DT(TFP2) and TF(L3) = DT(TFP3), see assumptions).

The first equation represents three modes of heat transfer relative to the pump inlet fluid volume (the volume within the wall node DT(TBN)).

1. Heat transfer due to mass transfer into the pump volume from the upstream line segment.

   $$\dot{M}Cp*(TF(L1)-DT(TFP1))$$

   where $\dot{M}Cp$ is equal to Q(L1)*RHOIL*CPFN

2. Convection to or from the pump walls around the inlet volume

   $$B1*(DT(TBN)-DT(TFP1))$$

   where B1 is equal to D(UP1B)*ASP1B, a convection coefficient.

3. Conduction to or from the upstream fluid line segment

   $$R1*(TF(L1)-DT(TFP1))$$

   R1 is the conduction coefficient equal to

   CF/(DXF(L1)/ACF(L1)+DXP1/ACP1+(RMFL1*DELT)/(ACF(L1)**2*RHOIL))

   where RMFL1=Q(L1)*RHOIL

These three heat transfer modes then combine to produce the heat balance equation for the pump inlet fluid node.

$$\frac{MCp}{DELT}*(DT(TFP1)-DT(TFP1)_{OLD}) = (R1+\dot{M}Cp)*(TF(L1)-DT(TFP1))+ B1*(DT(TBN)-DT(TFP1))$$

with MCp equal to FMASS1*CPFN

The second equation represents three modes of heat transfer relative to exit volume two (the volume within wall node DT(TCN) of the pump manifold).

1a. Convection to or from the piston node.

B3*(DT(TPN)-DT(TFP2))

where B3 is the convection coefficient equal to D(UP2P)*ASP2P.

1b. Convection to or from the pump walls at the exit chamber of the pump manifold, node DT(TCN)

B8*(DT(TCN)-DT(TFP2))

where B8 is equal to UP2C*ASP2C

2. Heat transfer due to mass transfer into the fluid volume from the inlet volume.

$\dot{M}Cp*(DT(TFP1)-DT(TFP2))$

where $\dot{M}Cp$ is equal to Q(L2)*RHOIL*CPFN

3. Heat added directly to the fluid due to compression, friction, and the piston moving parts.

D(HTREJ)*.323

where D(HIREJ) is defined in the Technical Summary.

These heat transfer modes are combined to produce the heat balanace equation for the pump exit fluid node.

$$\frac{MCp}{DELT}*(DT(TFP2)-DT(TFP2)_{OLD}) = B3*(DT(TPN)-DT(TFP1)) + \dot{M}Cp* (DT(TFP1)-DT(TFP2)) + .323 * D(HTREJ) +B8*(DT(TCN)-DT(TFP2)) \qquad (2)$$

The third equation represents three modes of heat transfer relative to volume three within the pump case.

1. Heat transfer due to mass transfer into the case volume three from the inlet, and exit volumes respectively. (Leakage flows).

DT(QLEAK1)*CPFN*(DT(TFP1)-DT(TFP3)) and

DT(QLEAK2)*CPFN*(DT(TFP2)-DT(TFP3))

where DT(QLEAK1) is equal to D(COECIN)*(P(L1)-P(L3)) and DT(QLEAK2) is equal to D(COEPLK)*(P(L2)-P(L3)).

2a. Convection to or from the pump mass node around the case.

B5*(DT(TDN)-DT(TFP3))

and B5 is equal to UP3D*ASP3D

2b. Convection to or from the piston mass node

B2*(DT(TPN)-DT(TFP3))

where B2 is equal to UP3D*ASP3P

3. Heat added to the fluid due to the heat rejection term

.24*D(HTREJ)

where D(HTREJ) was defined previously.

These heat transfer terms combine to produce the heat balance equation for the fluid volume 3 in the case drain.

$$\frac{MCp}{DELT}*(DT(TFP3)-DT(TFP3)_{OLD}) = DT(QLEAK1)*CPFN*(DT(TFP1)-DT(TFP3)) +DT(QLEAK2)*CPFN*(DT(TFP2)-DT(TFP3)) +B2*(DT(TPN)-DT(TFP3))+B5*(DT(TDN)-DT(TFP3)) + .24*D(HTREJ) \quad (3)$$

where MCp is equal to FMASS3*CPFN

The fourth equation represents four modes of heat transfer relative to the pump wall mass (inlet manifold mass) around the inlet volume.

1a. Conduction to or from the pump wall node around the exit volume.

(manifold node DT(TCN))

R9*(DT(TCN)-DT(TBN))

where R9 is equal to COB/(DXB/ACB+DXC/ACC)

1b. Conduction to or from the upstream line wall segment

R3*(TW(L1)-DT(TBN))

where R3 is the conduction coefficient equal to

1.0/(DXF(L1)/(ACW(L1)*C(L1))+DXB/(ACB*COB)

1c. Conduction to or from the pump walls around the case fluid volume.

R11*(DT(TDN)-DT(TBN))

where R11 is equal to COB/((DXB/ACB+DXD/ACD)*2.0)

1d. Conduction to or from the piston node

R5*(DT(TPN)-DT(TBN))

where R5 is equal to 1./(2.*(DXP/(D(ACP)*COP)+DXB/(ACB*COB)+

1.0/(D(ASPB)*D(CBP)))).

2a. Convection to or from the pump fluid in inlet volume, fluid

volume one.

B1*(DT(TFP1)-DT(TBN))

with B1 defined previously.

2b. Convection to or from the surrounding atmosphere

B6*(D(TA)-DT(TBN))

where B6 is equal to D(UAB)*D(ASAB)*D1

D1 is equal to D(VOL1)/(D(VOL1)+D(VOL2))

3. Heat added due to the heat rejection term

.125*D(HTREJ)

D(HTREJ) has been defined previously .

4. Radiation exchange with the surrounding structure

C2*(D(TST)-(DT(TBN)+460)**4)

where C2 is a radiation coefficient equal to C1*D1 where

C1 equals SIGMA*EPSION*SHAPF*D(ASAB) and D1 defined previously.

These heat transfer terms combine to produce the heat balance equation for the pump wall (manifold wall node B around the inlet volume).

$$\frac{MCp}{DELT}*(DT(TBN)-DT(TBN)_{OLD}) = R3+(TW(L1)-DT(TBN))+R9*(DT(TCN)-DT(TBN)) + R11*(DT(TDN)-DT(TBN)) + R5*(DT(TPN)-DT(TBN)) + B1*(DT(TFP1)-DT(TBN))+ B6*(DLTA)-DT(TBN)) + .125*D(HTREJ) +C2*(D(TST))-C2*(DT(TBN)+460.)**4 \quad (4)$$

where MCp is equal to D(TPMASS)*CPBN*D1.

The fifth equation represents three modes of heat transfer relative to the piston node.

1a. Convection to or from the fluid in the exit chamber

B3*(DT(TFP2)-DT(TPN))

with B3 described previously

1b. Convection to or from the case fluid

B2*(DT(TFP3)-DT(TPN))

with B2 being defined previously.

2. Conduction to or from the pump manifold walls

R5*(DT(TBN)-DT(TPN))

R8*(DT(TCN)-DT(TPN))

when R5 is as defined previously.

and R8 equals 1./((DXP/(D(ACP)*COP)+DXC/(ACC*COB)

+1./(D(ASBP)*D(CBP))*2.)

3. Heat added to the piston mass from the heat rejection term

.187*D(HTREJ)

These heat transfer terms combine to produce the heat balance equation for the piston node.

$$\frac{MCp}{DELT}*(DT(TPN)-DT(TPN)_{OLD}) = B3*(DT(TFP2)-DT(TPN)) + B2*(DT(TFP3)-DT(TPN)) + R5+(DT(TBN)-DT(TPN))+.187*D(HTREJ) +R8*(DT(TCN)-DT(TPN)) \quad (5)$$

where MCp is equal to D(PMASS)*CPPN

The sixth equation represents four modes of heat transfer relative to the pump manifold wall node surrounding the exit volume, Node C.

1a. Conduction to or from the downstream connecting line segment

R4*(TW(L2)-DT(TCN))

where R4 equals 1.0/(DXF(L2)/(ACW(L2)*C(L2)) + DXC/(ACC*COB))

1b. Conecution to or from the piston mass

R8*(DT(TPN)-DT(TCN))

where R8 was defined previously.

1c. Conduction to or from the two pump wall node manifold B (inlet, volume wall)

R9*(DT(TBN) - DT(TCN))

1d. Conduction to or from the case wall node

R10*(DT(TDN)-DT(TCN))

where R10 is equal to COB/((DXC/ACC+DXD/ACD)*2.)

and R9 was defined previously.

2a. Convection to or from the exiting fluid node

B8*(DT(TFP2)-DT(TCN))

where B8 was defined previously.

2b. Convection to or from the surrounding atmosphere

B9*(D(TA)-DT(TCN))

where B9 is equal to D(UAB)*D(ASAB)*D2

3. Heat added to the walls due to a heat rejection term,

.125*D(HTREJ)

where D(HTREJ) was defined previously.

4. Radiation exchange with the surrounding structure.

C3*(D(TST)-(DT(TCN)+460.)**4)

where C3 equals C1*D2 and C1 was defined previously, and D2 equals D(VOL2)/(D(VOL1)+D(VOL2)).

These heat transfer terms combine to produce the heat balance for the pump wall around the exit volume, manifold wall node (DT(TCN).

$$\frac{MCp}{DELT}*(DT(TCN)-DT(TCN)_{OLD}) = R4*(TW(L2)-DT(TCN))+R9*(DT(TBN)-DT(TCN))+R8*(DT(TPN)-DT(TCN))+.125*D(HTREJ)+R10*(DT(TDN)-DT(TCN)) + B9*(D(TA)-DT(TCN))+B8*(DT(TFP2)-DT(TCN)) +C3*D(TST)-C3*(DT(TCN)+460.)**4 \quad (6)$$

where MCp is equal to D(TPMASS)*CPBN*D2.

The seventh equation represents three modes of heat transfer relative to the pump walls surround the case fluid.

1a. Conduction to or from the case drain connecting line wall segment.

R7*(TW(L3)-DT(TDN))

where R7 is equal to 1.0/(DXF(L3)/(ACW(L3)*C(L3))+DXD/(ACD*COB))

1b. Conduction to or from the two other pump manifold wall nodes, around the inlet and outlet respectively.

R11*(DT(TBN)-DT(TDN))

R10*(DT(TCN)-DT(TDN))

where R10 and R11 were defined previously.

2a. Convection to or from the fluid in the case volume

B5*(DT(TFP3)-DT(TDN))

with B5 defined previously.

2b. Convection to or from the external surrounding atmosphere

B10*(D(TA)-DT(TDN))

where B10 is a convection coefficient equal to D(UAD)*D(ASAD).

3. Radiation exchange with the surrounding structure.

$$C4*(D(TST)-(DT(TDN)+460)^4)$$

where C4 is equal to SIGMA*EPSION*SHAPF*D(ASAD)

These heat transfer terms combine to produce the heat balance equation for the case wall node.

$$\frac{MCp}{DELT}*(DT(TDN)-DT(TDN)_{OLD}) = R7*(TW(L3)-DT(TDN))+R11*(DT(TBN)-DT(TDN)) + R10*(DT(TCN)-DT(TDN)) +B5*(DT(TFP3)-DT(TDN))+B10*(D(TA) -DT(TDN))+C4*D(TST)-C4*(DT(TDN)+460)**4 \tag{7}$$

where MCp is equal to D(PDMASS)*CPBN.

A thermal model of the above heat transfer terms for the pump is shown in Figure 6.51-3.

Equations (1) thru (7) are solved for the appropriate temperatures.

In the hydraulic math model the variable delivery pump generates fluid flow in response to system flow demand. The output pressure is a function of outlet flow. The steady state pump simulation models the pump characteristic flow versus pressure out curve (Figure 6.5-4), the characteristic leakage from high pressure to pump case, the leakage from pump case back to inlet and the pump outlet flow versus inlet pressure curve.

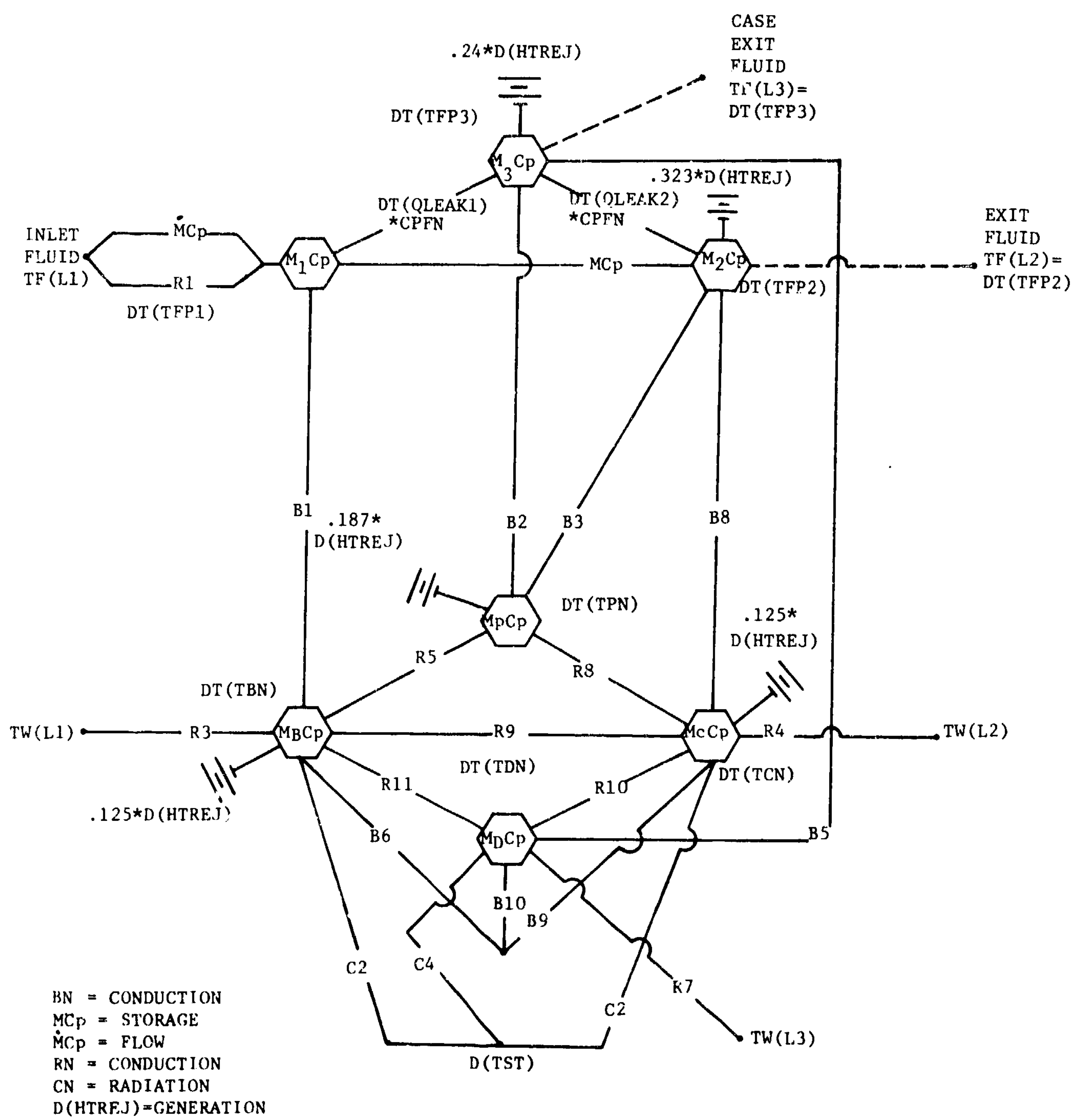


FIGURE 6.51-3

THERMAL MODEL

The inlet pressure is determined from the pump node. When inlet pressure falls below the minimum value, a warning message is printed.

On entry into the case drain section the leakage flow from high pressure to the pump case is calculated as

DT(QLEAK2) = DT(IRCDL)-(.3*(DT(QOUT)/DT(IRQ)

The flow that leaks back to the inlet from the case is a function of the case flow out the port and DT(QLEAK2). The case pressure is computed based on this flow difference.

DT(P_{CASE}) = D(RCDP)*(1.-DT(QCD)/DT(QLEAK2) + DT(PINLET)

Two equations are used to define the pump operating characteristics in Figure 6.51-4. When the outlet flow is less than DT(Q2R) the pump outlet pressure is calculated as

DT(POUTLT) = DT(IRPOQ) - (DT(IRPOQ)-DT(IRPFQ))*DT(QOUT)/DT(Q2R) (1)

If the outlet flow is greater than DT(Q2R)

DT(POUTLT) = DT(IRPFQ)*(1.-(DT(QOUT)-DT(Q2R))/(DT(Q3)-DT(Q2R))) (2)

The actual pump outlet pressure is calculated using DT(POUTLT) from the characteristic curve and adjusting this to account for the actual pressure in the case less the case pressure at which DT(POUTLT) was set.

DT(POUT) = DT(POUTLET) + DT(PCASE) - D(PSET)

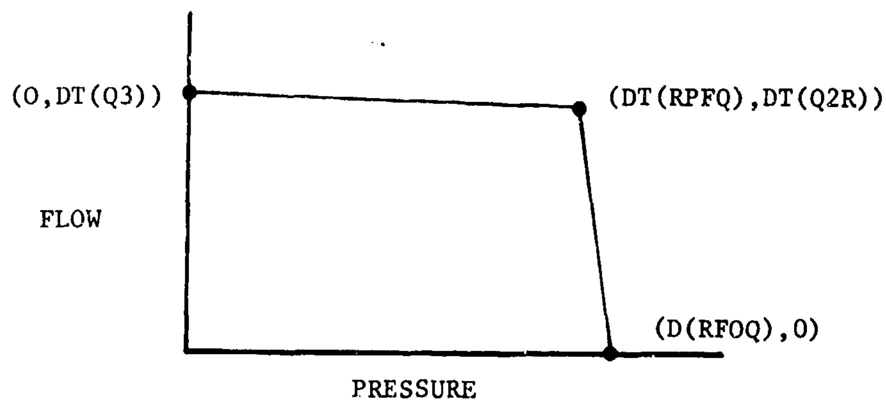


FIGURE 6.5-4

Pump Outlet Flow vs Outlet Pressure Curve

The IFLAGG indicator is set in TCALC to control which leg of the pressure/flow curve (Figure 6.51-4) to balance on. When IFLAGG is one, the outlet pressure is calculated using equation (1). IFLAGG equal to two selects equation (2) for computing the pump outlet pressure.

The IFLAGG indicator remains the same until a system flow balance is reached. Then the TCALC subroutine checks if the values are within the operating characteristics of the pump. When they are not, the appropriate IFLAGG indicator is set, and the iteration is restarted. Only one restart is allowed per time step.

6.51.2 Assumptions

1. All internal moving parts are evaluated as one node, all at the same temperature, DT(TPN).
2. The mass of the pump walls are modeled as three nodes, two top mainfold nodes one each associated with the inlet and exit fluid volumes, and the third wall around the case volume.
3. External temperatures remain constant.
4. Interface conductances between pump wall and connecting lines is infinite.
5. The fluids leaving volumes two (exit) and volume three(case) are equal to DT(TFP2) and DT(TFP3) respectively, so there is no interaction with the downstream line fluids nodes.
6. The emissivity of the walls remains constant, .3 for steel.
7. Complete mixing occurs in the fluid volumes.

6.51.3 Computational Methods

Section 1000

The fluid and wall temperatures are initialized, the external structure temperature is changed from degree Farenheit to Rankine and raised to the fourth power, and the default values are assigned.

The pump rated flow at the operating RPM is calculated as

DT(Q2R) = D(RQ)*D(RPM)/D(RPM)

The pump flow at zero system resistance is

DT(Q3) = 1.05*DT(Q2R)

Section 2000

The pump subroutine is called for each connection. The L(4) counter is incremented each time data is distributed to a connection. When L(4)=0 or 3 a new iteration has started and the pump values are calculated.

If a table of pump RPM values is input, INTERP will be called to determine the operating RPM at the current time step.

```
CALL INTERP(TIME1,D(33),D(L(9)),10,L(8),DT(ORPM),IERR)
```

The outlet flows and pressures are recomputed for the RPM.

Section 3000

Property values are assigned. Dimensions and coefficients are calculated. A 7 X 7 matrix is loaded and equations (1) through (7) are solved for DT(TFP1), DT(TFP2), DT(TFP3), DT(TBN), DT(TPN), DT(TCN) and DT(TDN). The calculated values are assigned to their proper storage locations and the boundary conditions are assigned to TF and TC in COMMON/TRANS/.

6.51.4 Approximations

1. The heat transfer coefficients for fluid in the case to the case walls is one third of the coefficient from fluid in volume one to the case walls.
2. Many distances and areas are approximated.

6.51.5 Limitations

The pump model is not valid for inlet pressure values less than the minimum inlet pressure.

6.51.6 Variable Listing

Variable	Description	Dimensions
ACB	Cross sectional of manifold wall node B, around the inlet volume	$IN.^2$
ACC	Cross Sectional area of manifold wall node C, around the exit volume	$IN.^2$
ACD	Cross sectional area of manifold wall node D, around the case volume	$IN.^2$
D(ACP)	Estimated cross-sectional area of the rotating group	$IN.^2$
ACP1	Estimated cross-sectional area of the inlet fluid	$IN.^2$

ACP2	Estimated cross-sectional area of the outlet fluid	$IN.^2$
ACP3	Estimated cross-sectional area of the case fluid	$IN.^2$
D(ASAB)	External surface area of the pump walls	$IN.^2$
D(ASPB)	Contact area, walls and the internal mass (pistons)	$IN.^2$
ASPIB	Surface area, inlet fluid to walls	$IN.^2$
ASP2P	Surface area, outlet fluid to pistons	$IN.^2$
ASP3P	Surface area, case fluid to walls	$IN.^2$
ASP3P	Surface area, case fluid to internal mass (pistons)	$IN.^2$
B	Dummy computational array	--
B1,B2,B3, B5, B6	Dummy variables	--
D(CPB)	Interface Conductance between the piston and walls	$WATTS/IN.^2 {}^\circ F$
CJ	Mechanical Equivalent of Heat	$FT\text{-}LB_m/WATTS\text{-}SEC$
COB	Thermal conductivity of the walls	WATTS/IN-°F
COP	Thermal conductivity of the pistons	WATTS/IN.-°F
CPBN	Specific heat of the walls	$WATTS\text{-}SEC/LB_m\text{-}{}^\circ F$
CPPN	Specific heat of the pistons	$WATTS\text{-}SEC/LB_m\text{-}{}^\circ F$
C1	Dummy variable	--
D(DELTA)	Distance from connection one to piston chamber	IN.
D(DELTA1)	Case Depth	IN.
DXB	Distance from wall node to interface of lines	IN.
DXC	Distance from internal fluid node to interface of lines	IN.
DXD	Distance from exit fluid node to interface	IN.
DXP	Distance from piston node to interface	IN.
DXP1	Distance from fluid one node to interface with line	IN.

6.51.6 Variable Listing (Continued)

Variable	Description	Dimensions
D1,D2	Dummy variables	--
EPSION	Emissivity factor	--
FMASS1	Inlet fluid mass	LB_m
FMASS2	Outlet fluid mass	LB_m
FMASS3	Case fluid mass	LB_m
D(HTREJ)	Heat rejection term	WATTS
D(ITF)	Initial temperature of the fluid in the pump	°F
D(ITB)	Initial temperature of the pump & piston masses	°F
ICOUNT	Number of balances in a time step	--
IFLAGG	Indicator for pump operating curve	--
LTYPE	Dummy variable	--
NTYPE	Dummy variable	--
DT(PCASE)	Case Pressure	PSI
DT(PINLET)	Inlet Pressure	PSI
D(PMASS)	Piston Mass (all internal moving parts)	LB_m
DT(POUTLT)	Outlet Pressure	PSI
POUT	Dummy Variable	-
PP	Computational array	-
D(PSET)	Pump Case Pressure at rated flow and pressure	PSI
D(PSMIN)	Minimum inlet pressure	PSI
D(PTYPE)	Piston Material Type	-
DT(QCD)	Case Drain Flow	CIS
DT(QLEAK1)	Leakage flow high pressure to case	CIS
DT(QLEAK2)	Leakage flow case to inlet	CIS

6.51.6 Variable Listing (Continued)

Variable	Description	Dimensions
DT(Q3)	Pump flow with zero system resistance	--
Q2C,QQQ	Dummy variables	--
DT(Q2R)	Rate Flow adjusted for operating RPM	CIS
D(RCDL), DT(IRCDL)	Case drain flow at rated conditions	CIS
D(RCDP)	Maximum pressure difference between pump case and inlet	PSID
RHOB	Case material density	LB_m/IN^3
RHOIL	Fluid density	LB_m/IN^3
RHOP	Rotating group material density	LB_m/IN^3
RMFL1, RMFL2, RMFL3	Dummy variables	--
D(RPFQ), DT(IRPFQ)	Rated pressure at full flow	PSI
D(RPM), DT(ORPM)	Pump operating speed	RPM
D(RPOQ), DT(IRPOQ)	Rated pressure at zero flow	PSI
D(RQ),DT(IRQ)	Rated flow	CIS
D(RRPM)	Pump speed at rated flow and pressure	RPM
RPM	Dummy variable	--
R1,R2, R3,R4, R5,R7, R10,R11	Dummy variables	--
SHAPF	Radiation shapf factor for the external walls	--
SIGMA	Stefan-Boltzmann radiation constant	$WATTS/IN.^2-°R^4$
D(TA)	Surrounding atmospheric temperature	°F
DT(TBN)	Temperature of the pump walls	°F

DT(TFP1)	Temperature of the inlet fluid	°F
DT(TFP2)	Temperature of the outlet fluid	°F
DT(TFP3)	Temperature of the case fluid	°F
D(TPMASS)	Pump wall mass	LB_m
DT(TPN)	Temperature of the internal parts, piston	°F
D(TST)	Temperature of the surrounding structure	°F
D(UAB)	External heat transfer coefficient of the pump	$WATTS/IN.^2$-°F
D(UP1B)	Heat transfer coefficient, inlet fluid to the walls	$WATTS/IN.^2$-°F
D(UP2P)	Heat transfer coefficient, outlet fluid and the piston	$WATTS/IN.^2$-°F
UP3P	Heat transfer coefficient, case fluid and the walls	$WATTS/IN.^2$-°F
D(VOL1)	Inlet volume	$IN.^3$
D(VOL2)	Outlet volume plus cylinders volume	$IN.^3$
D(VOL3)	Case volume	$IN.^3$

```
      SUBROUTINE TPUMP51 (D,DT,DD,L)
C**** REVISED DECEMBER 1979 ****
      DIMENSION D(64),DT(1),DD(1),L(10)
      COMMON G(90)
      COMMON /TRANS/P(300),Q(300),C(300),TC(300),TW(300),TF(300)
     + ,ACF(300),ACW(300),DXF(300),TIME,DELT,PI,NLINE,NEL
      COMMON /COMP/LTYPE(99),NC(99),KTEMP(99),IND,IENTR,INEL
      COMMON /STEADY/PN(90),QN(90),PEX(90),PDLEG(90),QL(90),
     + QA,QS,Q1,PUP,PDOWN,NNODE,NLEG,NCPN,TERM,LEGN,ICON,INV,
     + INX,INZ,NUP(90),NDWN(90),NELEM(90),ILEGAD(90),ILEG(1000),
     + IFLAGG,CON2,QQQ,ITER,ICOUNT
      COMMON /FLUID/ATPRES,CF,CPFN,FTEMP,PROP(13,3),IDAT,AT,IDSTT,STT,
     +IDWT,WT,IDTEMP,IDFLT,FLT
      COMMON/ENVIRON/NENV,VAT(5),VSTT(5),VWT(5)
      DIMENSION PP(7,7),B(7)
      INTEGER TST,TA,VOL1,VOL2,PMASS,VOL3,TPMASS,ACP,ASAB,TSTR,
     + UAB,ASPB,CBP,RPM,UP2P,DELTA,UP1B,QOUT
     + ,PTYPE,TFP1,TFP2,TFP3,TBN,TPN,QLEAK1,QLEAK2,PCASE
     + ,ASAD,PDMASS,TDN,TCN,RPFQ,RPOQ,RQ,RRPM,PSMIN,RCDP,
     + PSET,DELTA1,POUTLT,PINLET,Q3,Q2R,RCDL,HTREJ,QCD,POUT,
     + CON1,CON2,CON3,QIN,FLAG,ORPM
C       D ARRAY VARIABLES
      DATA MTYPE/1/,PTYPE/2/,TPMASS/3/,PMASS/4/,PDMASS/5/,VOL1/6/,
     + VOL2/7/,VOL3/8/,ACP/9/,ASPB/10/,UP2P/11/,HTREJ/12/,DELTA/13/,
     + ASAB/14/,ASAD/15/,UAB/16/,CBP/17/,UP1B/18/,TST/19/,
     + TA/20/,ITF/21/,ITB/22/,
     + RQ/23/,RRPM/24/,RPM/25/,RPOQ/26/,RPFQ/27/,
     + PSMIN/28/,RCDP/29/,RCDL/30/,PSET/31/,DELTA1/32/
C      DT ARRAY VARIABLES
      DATA TFP1/1/,TFP2/2/,TFP3/3/,TBN/4/,TPN/5/,QLEAK1/6/,QLEAK2/7/
     + ,Q2R/8/,Q3/9/,PINLET/10/,TCN/11/,TDN/12/,PCASE/13/,POUTLT/14/
     + ,QCD/15/,POUT/16/,QOUT/17/,QIN/18/,ORPM/19/,IRPOQ/20/,IRPFQ/21/
     + ,IRCDL/22/,IRQ/23/,TSTR/24/
      DATA SIGMA/.349E-11/,SHAPF/.96/,EPSION/0.3/,CJ/8.85/
C     DELTA  =THE DISTANCE FROM INLET TO OUTLET THROUGH THE PISTON
C     DELTA1 =THE TOTAL DEPTH OF THE DRAIN BOWL
C     ITF&ITB=INITIAL TEMPERATURE
C      UAB   =HEAT TRANSFER COEFF. ATMOSPHERE TO CASE
C     ASAD  =SURFACE AREA EXTERNAL TO CASE DRAIN WALL
C     PDMASS =PUMP WALL MASS OF CASE DRAIN WALLS
C      CBP    =INTERFACE CONDUCTANCE, PISTON TO CASE
C     VOL1   =INLET VOLUME
C     VOL2      =EXIT VOLUME INCLUDES CYLINDER VOLUMES
C     VOL3    =CASE VOLUME
C     ASPB =CONTACT AREA INTERNAL PARTS,PISTON,TO CASE
C      ACP   =CROSS SECTIONAL AREA OF TOTAL PISTON, INTERNAL
C      MASS
C      UP2P =HEAT TRANSFER COEFFICIENT PISTON TO EXIT FLUID
C     QLEAK1    =LEAKAGE FLOW FROM INLET TO DRAIN
C     UP1B      =HEAT TRANSFER COEFF. CASE OR PISTON TO FLUIDS
```

```
C      PMASS   =PISTON MASS
C      TPMASS =PUMP MASS SURROUNDING VOLUMES 1& 2,TOP MOUNTING
C      320 BTU/MIN.=5625.WATTS
       IF(IENTR)1000,2000,3000
C ***  1000 SECTION
 1000  CONTINUE
       IF(IDTEMP .EQ. 0 .OR. L(9) .EQ. 0) GOTO 1100
       IF(IDAT .EQ. 1) D(20)=AT
       IF(IDSTT .EQ. 1) D(19)=STT
       IF(IDFLT .EQ. 1) D(21)=FLT
       IF(IDWT .EQ. 1) D(22)=WT
 1100  IF(L(10).EQ.0)GOTO 333
C ***  SET DATA TO PROPER ENVIRONMENT ***
       D(TA)=VAT(L(10))
       D(TST)=VSTT(L(10))
       D(ITB)=VWT(L(10))
C
  333  DO 20 I=1,30
   20  DT(I)=0.0
       L1=L(1)
       L2=L(2)
       L3=L(3)
C      INITIALIZING TEMPERATURES
       TC(L1)=D(ITB)
       TC(L3)=D(ITB)
       TF(L3)=D(ITF)
       TC(L2)=D(ITB)
       TF(L2)=D(ITF)
       TF(L1)=D(ITF)
       DT(TFP1)=D(ITF)
       DT(TCN)=D(ITB)
       DT(TDN)=D(ITB)
       DT(TFP2)=D(ITF)
       DT(TFP3)=D(ITF)
       DT(TBN)=D(ITB)
       DT(TPN)=D(ITF)
       DT(TSTR)=(D(TST)+460.)**4
       IF(D(UP1B).EQ.0.0) D(UP1B)=.1
       IF(D(UAB).EQ.0.0) D(UAB)=.0069
       IF(D(UP2P).EQ.0.0) D(UP2P)=3.0
       L(4)=0
       DT(Q2R)=D(RQ)*D(RPM)/D(RRPM)
       DT(Q3)=1.05*DT(Q2R)
       QQQ=DT(Q2R)
C      IF (PN(NDWN(L(5))).EQ.0.0)PN(NDWN(L(5)))=48.
       DT(PCASE)=20.
       G(L(7))=2.
       LL=(L(8)+7)/8
       L(9)=33+LL*8
       IERR=0
       DT(IRPFQ)=D(RPFQ)
       DT(IRPOQ)=D(RPOQ)
       DT(IRCDL)=D(RCDL)
       DT(IRQ)=DT(Q2R)
       RETURN
```

```
C *** 2000 SECTION
 2000 IF (L(4) .EQ. 3) L(4)=0
      IF (L(4) .NE. 0) GO TO 2100
      CON1=L(5)
      CON2=L(6)
      CON3=L(7)
      IF (L(8).GT.0)GO TO 2010
C     SET INLET AND OUTLET PRESSURES
 2020 DT(PINLET)=PN(NDWN(CON1))
      IF(DT(PINLET).LT.D(PSMIN).AND.TIME.NE.0.0)WRITE(6,999)D(PSMIN)
  999 FORMAT(10X,34HWARNING PUMP INLET PRESSURE BELOW ,F10.2,4H PSI)
      DT(QOUT)=QL(CON2)
      IF (DT(QOUT).LT.0.)DT(QOUT)=0.
      IF(DT(QOUT).GT.DT(Q3))DT(QOUT)=DT(Q3)
      IF (IFLAGG .EQ. 2)GO TO 2001
      DT(POUTLT)=DT(IRPOQ)-(DT(IRPOQ)-DT(IRPFQ))*DT(QOUT)/DT(Q2R)
      GO TO 2002
 2001 DT(POUTLT)=DT(IRPFQ)*(1.-(DT(QOUT)-DT(Q2R))/(DT(Q3)-DT(Q2R)))
C     CALCULATE LEAKAGE FLOW FROM PRESSURE TO CASE
 2002 DT(QLEAK2)=DT(IRCDL)-(.3*DT(QOUT)/DT(IRQ))
C     READ CASE FLOW AND CHECK AGAINST LIMITS
      DT(QCD)=QL(CON3)
      IF (DT(QCD).GT.DT(QLEAK2))DT(QCD)=DT(QLEAK2)
      IF (DT(QCD).LT.0.)DT(QCD)=0.0
C     CALCULATE DP BETWEEN CASE AND INLET, SET CASE PRESSURE
      DPCS=D(RCDP)*(1.-DT(QCD)/DT(QLEAK2))
      DT(PCASE)=DT(PINLET)+DPCS
      DT(QIN)=DT(QOUT)+DT(QCD)
      GO TO 2100
C *** VARIABLE RPM SECTION
 2010 TIME1=TIME-1/3*DELT
      CALL INTERP(TIME1,D(33),D(L(9)),10,L(8),DT(ORPM),IERR)
C     ADJUST OUTLET FLOWS AND PRESSURES FOR RPM
C        DT(ORPM)  =  OPERATING RPM
      DT(Q2R)=D(RQ)*DT(ORPM)/D(RRPM)
      DT(Q3)=1.05*DT(Q2R)
      DT(IRPOQ)=D(RPOQ)*DT(ORPM)/D(RRPM)
      DT(IRPFQ)=D(RPFQ)*DT(ORPM)/D(RRPM)
      QQQ=DT(Q2R)
      DT(IRCDL)=D(RCDL)*DT(ORPM)/D(RRPM)
      DT(IRQ)=D(RQ)*DT(ORPM)/D(RRPM)
      IF(ITER.EQ.1.AND.ICOUNT.EQ.1)IFLAGG=1
      IF (DT(ORPM).GT.0.0)GO TO 2015
C     IF RPM=0, SET PARAMETERS TO ZERO FLOW VALUES
      DT(PINLET)=PN(NDWN(CON1))
      DT(QOUT)=0.0
      DT(POUTLT)=D(PSET)
      DT(QLEAK2)=0.0
      DT(QCD)=0.0
      DT(PCASE)=DT(PINLET)
      DT(QIN)=0.0
      QQQ=0.1
      GO TO 2100
C     MAKE NEW FLOW GUESSES BASED UPON OPERATING RPM
 2015 IF (QL(CON1).NE.0.0.AND.QL(CON2).NE.0.0)GO TO 2020
```

```
      QL(CON1)=DT(Q2R)
      QL(CON2)=0.9*DT(Q2R)
      QL(CON3)=2.
      GO TO 2020
C *** DISTRIBUTE DATA TO CONNECTIONS
 2100 IF (ICON-2)2200,2300,2400
C     INLET
 2200 Q1=QL(INEL)
      L(4)=L(4)+1
C     FLAG=0
      RETURN
C     OUTLET
 2300 IF (INX.NE.1) GO TO 2700
      Q1=DT(QOUT)
      DT(POUT)=DT(POUTLT)+DT(PCASE)-D(PSET)
      IF(DT(POUT).LT.DT(PINLET))DT(POUT)=DT(PINLET)
      PUP=DT(POUT)
      PDLEG(INEL)=DT(POUT)-DT(PINLET)
      INV=0
      TERM=PDLEG(INEL)
      L(4)=L(4)+1
      RETURN
C     CASE DRAIN
 2400 IF (INX NE.1) GO TO 2700
      Q1=DT(QCD)
      PDLEG(INEL)=PN(NDWN(INEL))-PN(NUP(INEL))+DT(QCD)/G(INEL)
      TERM=PDLEG(INEL)
      DT(QLEAK1)=DT(QLEAK2)-DT(QCD)
      L(4)=L(4)+1
      PUP=PDLEG(INEL)+DT(PINLET)
      INV=0
      RETURN
 2700 WRITE(6,2800) IND,ICON,INEL
 2800 FORMAT(5X,46H CALL SEQUENCE ERROR DETECTED IN COMPONENT NO ,
     + I5,14H CONNECTION NO,I5,7HLEG NO.,I5)
      STOP 5000
C *** 3000 SECTION
 3000 CONTINUE
      L1=L(1)
      L2=L(2)
      L3=L(3)
      KTYPE=D(NTYPE)+.001
      NTYPE=D(PTYPE)+.001
      CPBN=PROP(KTYPE,1)
      CPPN=PROP(NTYPE,1)
      COB=PROP(KTYPE,3)
      COP=PROP(NTYPE,3)
      CPCN=CPBN
      CPDN=CPBN
      RHOP=PROP(NTYPE,2)
      RHOB=PROP(KTYPE,3)
C     AREAS & DISTANCES ARE ESTIMATES
      D1=D(VOL1)/(D(VOL1)+D(VOL2))
      D2=D(VOL2)/(D(VOL1)+D(VOL2))
      DXP1=D(DELTA)/4.0
```

```
      DXB=D(DELTA)/4.0
      DXC=D(DELTA)/4.0
      DXD=D(DELTA1)/2.
      DXP=D(PMASS)/(RHOP*D(ACP))
      C1=SIGMA*EPSION*SHAPF*D(ASAB)
      C2=C1*D1
      C3=C1*D2
      C4=SIGMA*EPSION*SHAPF*D(ASAD)
      RHOIL=386.4*RHO(DT(TFP1),(P(L3)+P(L1))/2.)
      FMASS1=D(VOL1)*RHOIL
      FMASS2=(D(VOL2))*RHOIL
      FMASS3=D(VOL3)*RHOIL
      CMASS=D(TPMASS)*D2
      BMASS=D(TPMASS)*D1
      ACB=BMASS/(RHOB*D(DELTA)/2.)
      ACC=CMASS/(RHOB*D(DELTA)/2.)
      ACPP=D(PMASS)/RHOP
      ACD=D(PDMASS)/(RHOB*D(DELTA1))
      ACP1=D(VOL1)/D(DELTA)
      ACP2=D(VOL2)/(D(DELTA)/2.0)
      ACP3=D(VOL3)/(D(DELTA1)/2.0)
      ASP1B=SQRT(4.*ACP1/PI)*PI*D(DELTA)
      ASP2P=1.3*SQRT(4.*ACP2/PI)*PI*D(DELTA)/2.
      ASP2C=SQRT(4.*ACP2/PI)*PI*D(DELTA)/2.
      ASP3P=1.39*SQRT(4.*ACPP/PI)*PI*D(DELTA1)
      ASP3D=.75*D(ASAD)
C     HEAT TRANSFER COEFF.ARE CONSTANTS=50BTU/HR-FT2-F(.11 WATTS
C     /IN2-F) DEFAULT VALUE
      UP3D=D(UP1B)/3.
      UAD=D(UAB)
      UP2C=D(UP1B)/1.
      RMFL1=ABS(Q(L1))*RHOIL
      RMFL2=ABS(Q(L2))*RHOIL
      RMFL3=+ABS(Q(L3))*RHOIL
 3200 R1=CF/(DXF(L1)/ACF(L1)+DXP1/ACP1+(RMFL1*DELT)/(ACF(L1)
     + **2*RHOIL))
      R3=1.0/(DXF(L1)/(ACW(L1)*C(L1))+DXB/(ACB*COB))
      R4=1.0/(DXF(L2)/(ACW(L2)*C(L2))+DXC/(ACC*COB))
      R5=1.0/((DXP/(D(ACP)*COP)+DXB/(ACB*COB)+
     + 1./(D(ASPB)*D(CBP)))*2.)
      R7=1.0/(DXF(L3)/(ACW(L3)*C(L3))+DXD/(ACD*COB))
      R9=COB/(DXB/ACB+DXC/ACC)
      R10=COB/((DXC/ACC+DXD/ACD)*2.)
      R11=COB/((DXB/ACB+DXD/ACD)*2.)
      R8=1./((DXP/(D(ACP)*COP)+DXC/(ACC*COB)+1./(D(ASPB)*D(CBP)))*2.)
      B1=D(UP1B)*ASP1B
      B2=UP3D*ASP3P
      B3=D(UP2P)*ASP2P
      B5=UP3D*ASP3D
      B6=D(UAB)*D(ASAB)*D1
      B8=UP2C*ASP2C
      B9=D(UAB)*D(ASAB)*D2
      B10=UAD*D(ASAD)
C      P1,P2 P3 B,P,C,D, NODES IN ORDER
```

```
      DO 3333 J=1,7
      PP(I,J)=0.0
 3333 B(I)=0.0
 3300 PP(1,1)=FMASS1*CPFN/DELT+R1+CPFN*RMFL1+B1
     + +DT(QLEAK1)*RHOIL*CPFN
      PP(1,3)=-DT(QLEAK1)*RHOIL*CPFN
      PP(1,4)=-B1
      B(1)=FMASS1*CPFN*DT(TFP1)/DELT+(R1+CPFN*RMFL1)*TF(L1)
      PP(2,1)=-(RMFL1+DT(QLEAK1)*RHOIL)*CPFN
      PP(2,2)=FMASS2*CPFN/DELT+(RMFL1+DT(QLEAK1)*RHOIL)*CPFN+B3+B8
      PP(2,6)=-B8
      PP(2,5)=-B3
      B(2)=FMASS2*CPFN*DT(TFP2)/DELT+0.30*D(HTREJ)
      PP(3,2)=-DT(QLEAK2)*CPFN*RHOIL
      PP(3,3)=FMASS3*CPFN/DELT+B5+DT(QLEAK2)*CPFN*RHOIL+B2
      PP(3,7)=-B5
      PP(3,5)=-B2
      B(3)=FMASS3*CPFN*DT(TFP3)/DELT+0.10*D(HTREJ)
      PP(4,1)=-B1
      PP(4,4)=BMASS*CPBN/DELT+R9+R3+R5+B1+B6+R11
      PP(4,6)=-R9
      PP(4,7)=-R11
      PP(4,5)=-R5
      B(4)=BMASS*CPBN*DT(TBN)/DELT+B6*D(TA)+R3*TW(L1)+
     + C2*DT(TSTR)-C2*(DT(TBN)+460.)**4
      PP(5,6)=-R8
      PP(5,2)=-B3
      PP(5,3)=-B2
      PP(5,4)=-R5
      PP(5,5)=D(PMASS)*CPPN/DELT+B3+R5+B2+R8
      B(5)=D(PMASS)*CPPN*DT(TPN)/DELT+0.60*D(HTREJ)
      PP(6,4)=-R9
      PP(6,5)=-R8
      PP(6,7)=-R10
      PP(6,2)=-B8
      PP(6,6)=CMASS*CPBN/DELT+R4+R8+R9+R10+B8+B9
      B(6)=CMASS*CPBN*DT(TCN)/DELT+R4*TW(L2)+B9*D(TA)+C3*DT(TSTR)
     + -C3*(DT(TCN)+460.)**4
      PP(7,3)=-B5
      PP(7,4)=-R11
      PP(7,6)=-R10
      PP(7,7)=D(PDMASS)*CPBN/DELT+R7+R10+R11+B5+B10
      B(7)=D(PDMASS)*CPBN*DT(TDN)/DELT+R7*TW(L3)+B10*D(TA)
     + +C4*DT(TSTR)-C4*(DT(TDN)+460.)**4+.001*D(HTREJ)
 3600 CALL SIMULT(PP,B,7,IERROR)
      DT(TFP1)=B(1)
      DT(TFP2)=B(2)
      DT(TFP3)=B(3)
      DT(TBN)=B(4)
      DT(TCN)=B(6)
      DT(TDN)=B(7)
      DT(TPN)=B(5)
      TF(L2)=B(2)
      TF(L3)=B(3)
      TC(L1)=B(4)
      TC(L2)=B(6)
      TC(L3)=B(7)
      RETURN
      END
```

APPENDIX T
CALCULATION SUBROUTINES
SUBROUTINE TCALC
HYTTHA TECHNICAL MANUAL (AFAPL-TR-76-43, VOL. VIII)

4.2 SUBROUTINE TCALC

The TCALC subroutine is responsible for the steady state calculations in the system. TCALC is called from the THYTR main program. The subroutine will compute the pressures at all the system nodes and flows in all the legs, using pressure drop data obtained from TLEGCAL. Figure 4.2-1 is a generalized flow diagram of TCALC.

On entry into TCALC the first phase performed by the subroutine will be to initialize the appropriate calculation arrays. After the initialization, the computation phase begins. All the legs will be assigned conductance values from the TLEGCAL subroutine. These conductance values, along with constant factors, will then be inserted into two matrices. The TGAUSS subroutine will be called to compute the new pressure values. These pressure values at the nodes are then used to calculate the new flow rates for the legs in the system. When all the flows pass the convergence test, the flows and pressures are written to labeled common arrays and program control is passed back to THYTR. If the number of iterations exceeds 50, the most recent calculated values of flow and pressure are returned to the labeled common arrays and an error message is printed.

4.2.1 Math Model - The development of the TCALC subroutine to analyze complex flow systems results from the assumption that all resistance factors in a line can temporarily be assumed linear. The net flow around any node can then be written as the sum of all the flows entering and leaving that node or $Q_{NET} = 0$.

If R_{12} is a resistance factor used to describe a resistance in a leg, then $R_{12} = \Delta P_{12}/Q_{12}$.

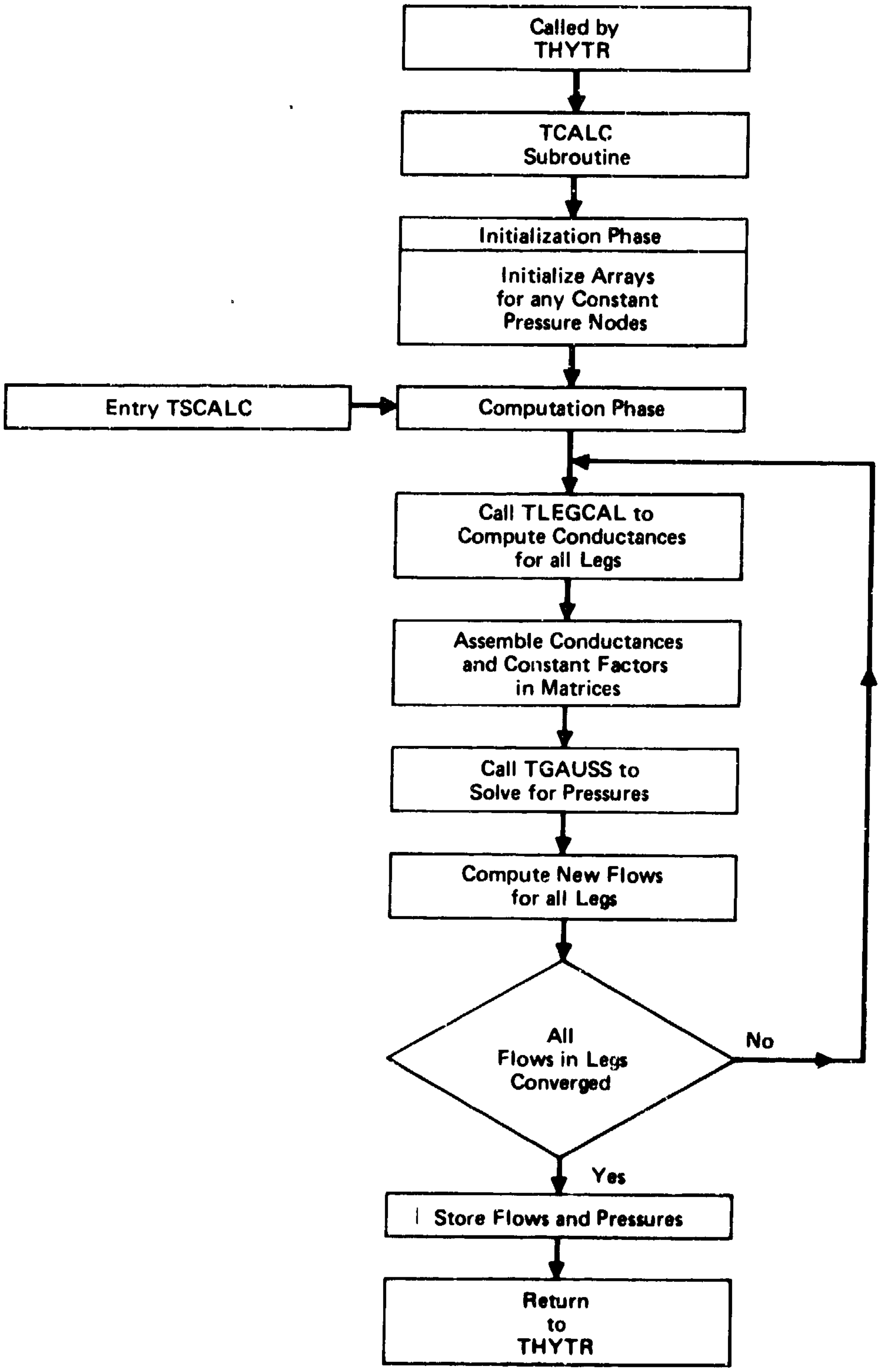


FIGURE 4.2-1
TCALC GENERALIZED FLOW DIAGRAM

where:

R_{12} = Resistance from node 1 to node 2 of the leg

ΔP_{12} = Pressure drop from node 1 to node 2 of the leg

Q_{12} = Flow in the leg

Conductance is then defined as:

$$G_{12} = \frac{1}{R_{12}}$$

where:

G_{12} = Conductance from node 1 to node 2 of the leg

Then:

$$Q_{12} = G_{12} P_{12}$$

The net flow at any node (where three or more legs come together) must be zero.

Therefore, the flow requirement is satisfied if:

$$\Sigma_J G_{IJ} [P_I - P_J \pm \Delta P_{IJ}] - \Sigma_K \pm Q_{IK} = 0$$

Where:

P_I = pressure at node I

P_J = pressure at node J

ΔP_{IJ} = a pressure rise or loss (from a pump or actuator) in leg IJ

Q_{IK} = fixed flow in leg IK connected to node I

Equations of the above form are input to a matrix for solution of pressures at nodes. These matrix solution pressures are used in conjunction with the calculated conductance (G) to calculate a new flow guess in each leg. When two successive flow guesses for all legs in the system are within a specific tolerance such as .001 CIS, the solution has converged. Refer to Appendix A SSFAN Technical Manual, AFAPL-TR-76-43, Vol. VI, for a more detailed mathematical development.

4.2.2 TCALC Subroutine Description - The TCALC subroutine is divided into two phases. The first phase deals directly with the input data for establishing the system pressure node identification arrays. Six arrays are generated which are used in the calculation of node pressures and leg flows in phase two. Specifically, these arrays are:

JCOL:

1) Dimension (M,20)

2) The final JCOL array (in compressed form) identifies the columns in a square CALC1 array which are filled with non-zero terms. The rows of JCOL correspond to the rows of CALC1, and the elements in each row of JCOL correspond to the column number in each row of CALC1.

3) Note: JCOL describes a square CALC1 array in order to be compatible with the solution technique in TGAUSS.

IDIAG:

1) Dimension(M)

2) The IDIAG array identifies which columns of CALC1 contain the positive flow values. IDIAG(1) corresponds to the column in which the positive conductance is located in the first row of the CALC1 array. IDIAG(2) corresponds to the column in which the positive conductance is located in the second row of the CALC1 array.

3) Note: IDIAG describes a compressed CALC array.

JNEG:

1) Dimension (ML)

2) The JNEG array identifies which column in CALC1 contains the first appearance (in a row-by-row search) of a negative conductance value in the CALC1 array. For example, JNEG(4) records the first negative conductance value of LEG4. If JNEG(4)=3, then the first time a $-G_4$ appears is in Row 4, Column 3 of the CALC1 array.

3) Note: JNEG describes a compressed CALC1 array.

INEG:

1) Dimension (ML)

2) The INEG array differs from the JNEG array in only one respect, that being the INEG array stores the second appearance of a negative conductance value.

3) Note: The INEG array describes a compressed CALC1 array.

JRENT:

1) Dimension(M)

2) The JRENT array identifies the number of non-zero entries in each row of CALC1.

3) Note: JRENT describes either a square or compressed CALC1 array.

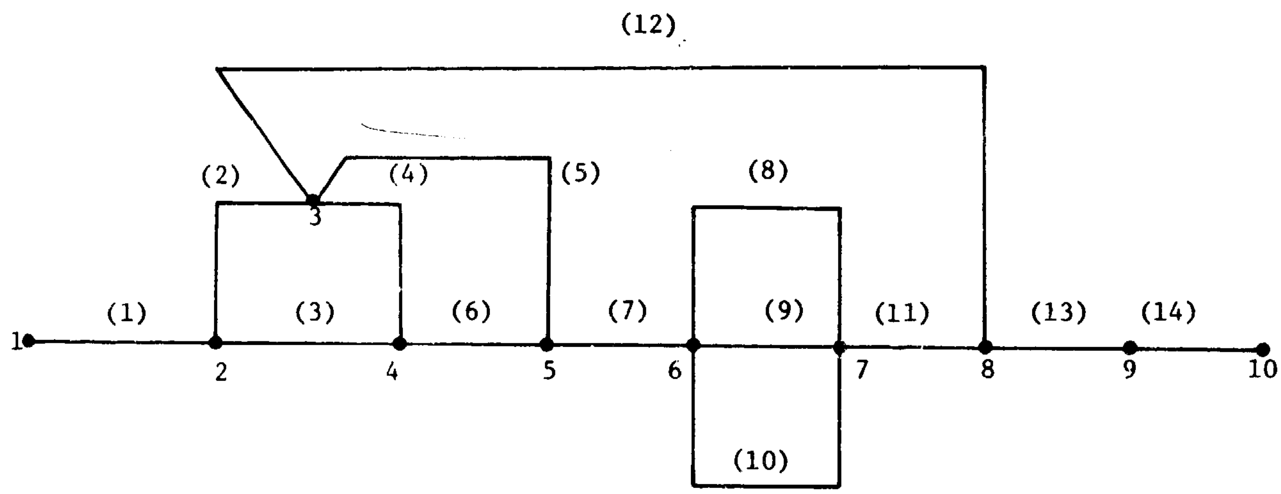
JCENT:

1) Dimension(M)

2) The JCENT array identifies the number of non-zero entries in each column of CALC1.

3) Note: JCENT describes a <u>square</u> CALC1.

To understand how Phase I works, the example system in Figure 4.2-2 is developed below. A simplified flow diagram of Phase I is shown in Figure 4.2-3.



1	1	2
2	2	3
3	2	4
4	3	4
5	3	5
6	4	5
7	5	6
8	6	7
9	6	7
10	6	7
11	7	8
12	3	8
13	8	9
14	9	10

ILEP ARRAY

NUMBER OF PRESSURE NODES M = 10

NUMBER OF LEGS ML = 14

Figure 4.2-2

TCALC EXAMPLE SYSTEM

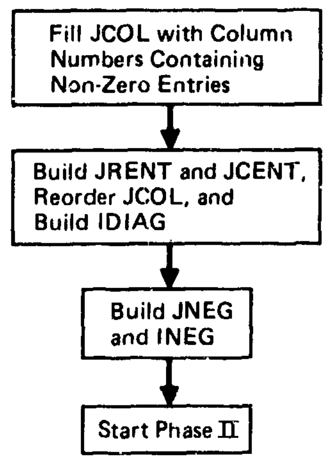


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FIGURE 4.2-3
TCALC SUBROUTINE PHASE ONE OPERATION

I. JCOL is initially filled with non-zero terms to indicate the positions of non-zero terms in a square CALC1.

```
DO 10 K=1, ML
I = ILEF(K,2)
J = ILEP(K,3)
JCOL(I,J)=I
JCOL(J,I)=I
JCOL(I,I)=I
JCOL(J,J)=I
```

JCOL =

	1	2	3	4	5	6	7	8	9	10
1	1	1								
2	1	3	2	3						
3		2	12	4	5			12		
4		3	4	6	6					
5			5	6	7	7				
6					7	10	10			
7						10	11	11		
8			12				11	13	13	
9								13	14	14
10									14	14

II. JCOL is renumbered to provide for easy construction of JRENT, JCENT, JNEG, INEG, and IDIAG.

```
C-----RENUMBER JCOL AND BUILD JRENT
      DO 20 I=1,M
      KOUNT=0
      DO 35, J=1,M
      JJ=JCOL(I,J)
      IF(JJ.EQ. ) GO TO 35
      KOUNT=KOUNT+1
      JCOL (I,J)=KOUNT
 35   CONTINUE
      JRENT(I)=KOUNT
 20   CONTINUE
```

JCOL =

	1	2	3	4	5	6	7	8	9	10
1	1	2								
2	1	2	3	4						
3		1	2	3	4			5		
4		1	2	3	4					
5			1	2	3	4				
6					1	2	3			
7						1	2	3		
8			1				2	3	4	
9								1	2	3
10									1	2
	1	2	3	4	5	6	7	8	9	10

JRENT =

2	4	5	4	4	3	3	4	3	2

III. JNEG records the CALC1 column containing the downstream appearance of the leg number as a negative element. INEG records its upstream occurrence.

```
C ------ LOCATE ALL OF OFF-DIAGONAL ELEMENTS
         DO 45 K=1,ML
         I=ILEP(K,2)
         J=ILEP(K,3)
         JNEG(K)=JCOL(I,J)
   45    INEG(K)=JCOL(J,I)
```

	1	2	3	4	5	6	7	8	9	10	11	12	13	14
JNEG =	2	3	4	3	4	4	4	3	3	3	3	5	4	3

	1	2	3	4	5	6	7	8	9	10	11	12	13	14
INEG =	1	1	1	2	1	2	1	1	1	1	2	1	1	1

IV.

```
C ------ BUILD JCENT AND IDIAG
         DO 65 I=1,M
         IDIAG(K)=JCOL(K,K)
         KOUNT=0
         DO 67 J=1,M
         IF(JCOL(J,I).EQ.0)GO TO 67
         KOUNT=KOUNT+1
   67    CONTINUE
   65    JCENT(I)=KOUNT
```

	1	2	3	4	5	6	7	8	9	10
JCENT =	2	4	5	4	4	3	3	4	3	2

	1	2	3	4	5	6	7	8	9	10
IDIAG =	1	2	2	3	3	2	2	3	2	2

V. The JCOL elements are all left-justified, and their previous positions are set equal to zero by the statement JCOL(I,J)=0. This statement must precede JCOL(I,K)=J so that, in the event that J=K, the compressed JCOL matrix contains its non-zero elements in the proper location. ICOL can now be copied from JCOL and be passed to GAUSS for use in the solution process.

```
C ------ COMPRESS THE JCOL MATRIX
         DO 70 I=1,M
         NN =JRENT(I)
         J=0
         DO 70 K=1,NN
   75    J=J+1
         K1=JCOL(I,J)
         IF(K1.EQ.0)GO TO 75
         JCOL(I,J)=0
         JCOL(I,K)=J
   70    CONTINUE
```

$$
JCOL = \begin{array}{c|ccccc|}
 & 1 & 2 & 3 & 4 & 5 \\
\hline
1 & 1 & 2 & & & \\
2 & 1 & 2 & 3 & 4 & \\
3 & 2 & 3 & 4 & 5 & 8 \\
4 & 2 & 3 & 4 & 5 & \\
5 & 3 & 4 & 5 & 6 & \\
6 & 5 & 6 & 7 & & \\
7 & 6 & 7 & 8 & & \\
8 & 3 & 7 & 8 & 9 & \\
9 & 8 & 9 & 10 & & \\
10 & 9 & 10 & & & \\
\hline
\end{array}
$$

Phase two operation of the CALC subroutine begins with initializing the conductance array - CALC1, and the constant array - CALC2, to zero values. (See Figure 4.2-4 for a flow diagram of the phase two operation.) A call is now made to the subroutine TLEGCAL for each leg in the system. TLEGCAL will return the value of conductance to the G array in the unlabeled common.

After all the conductance values are calculated for each leg, they must be entered into the compressed CALC1 array. For the example system the CALC1 array contains:

```
C ------ BUILD CALC1 MATRIX
         DO 9099 K=1,ML
         I=ILEP(K,2)
         J=ILEP(K,3)
         L=IDIAG(I)
         LM=IDIAG(J)
         CALC1(I,L)=CALC1(I,L)+G(K)
         CALC1(J,LM)=CALC1(J,LM)+G(K)
         L=JNEG(K)
         LM=INEG(K)
         CALC1(I,L)=CALC1(I,L)-G(K)
         CALC1(J,LM)=CALC1(J,LM)-G(K)
9099     CONTINUE
```

	1	2	3	4	5
1	G_1	$-G_1$	0	0	0
2	$-G_1$	$G_1+G_2+G_3$	$-G_2$	$-G_3$	0
3	$-G_2$	$G_2+G_4+G_5+G_{12}$	$-G_4$	$-G_5$	$-G_{12}$
4	$-G_3$	$-G_4$	$G_3+G_4+G_6$	$-G_6$	0
5	$-G_5$	$-G_6$	$G_5+G_6+G_7$	$-G_7$	0
6	$-G_7$	$G_7+G_8+G_9+G_{10}$	$G_8-G_9-G_{10}$	0	0
7	$-G_8-G_9-G_{10}$	$G_8+G_9+G_{10}+G_{11}$	$-G_{11}$	0	0
8	$-G_{12}$	$-G_{11}$	$G_{11}+G_{12}+G_{13}$	$-G_{13}$	0
9	$-G_{13}$	$G_{13}+G_{14}$	$-G_{14}$	0	0
10	$-G_{14}$	G_{14}	0	0	0
	1	2	3	4	5

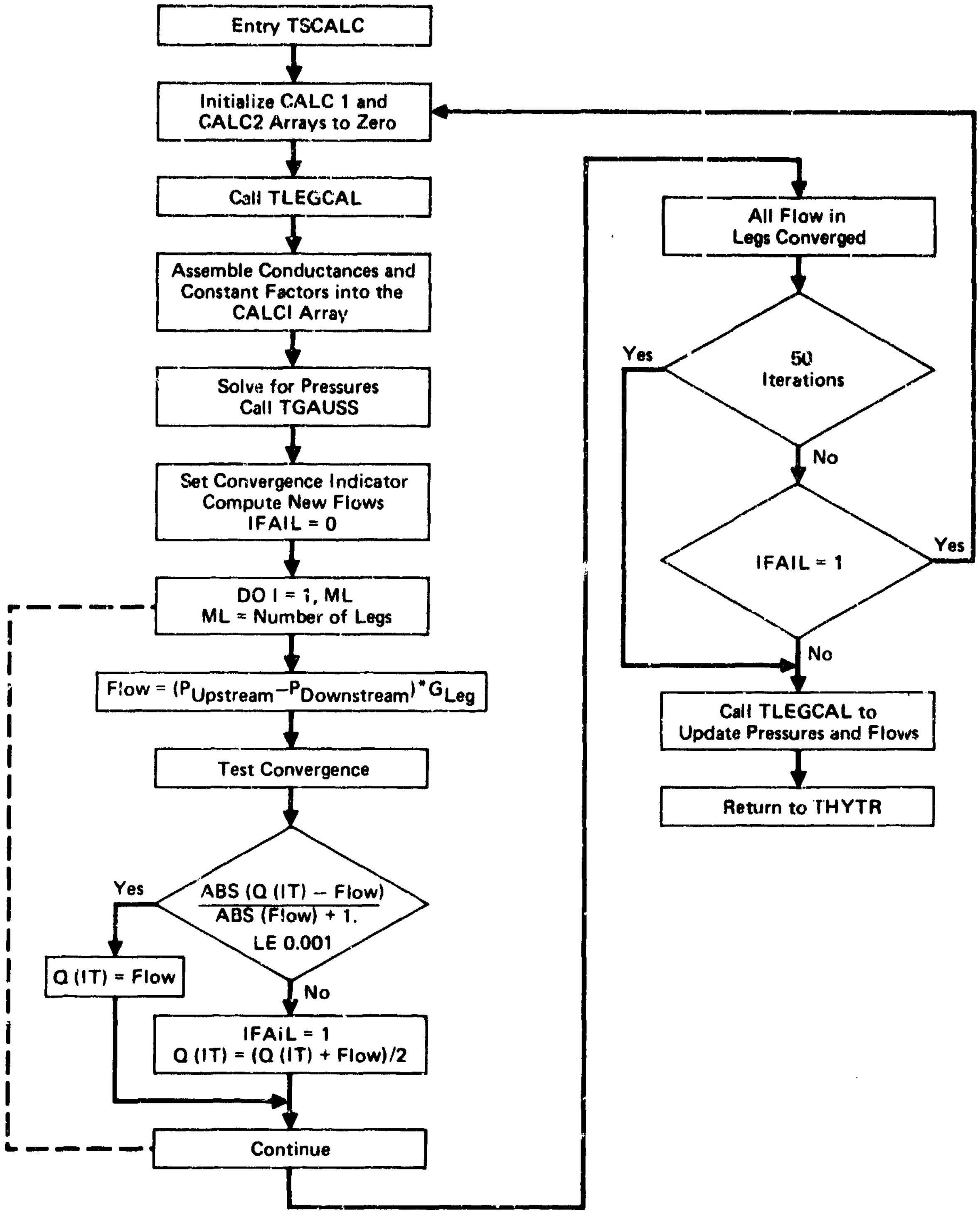

FIGURE 4.2-4
CALC SUBROUTINE PHASE TWO OPERATION

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CALC1 is built in this manner for each iteration. This compressed form speeds the solution process.

The CALC2 array contains the constant terms of the system of linear equations that describe the model. Constant pressure drops in legs, external flows and constant pressure sources are all inserted into this array. Any constant pressure source or pressure drop is multiplied by the conductance of the leg it is associated with. If leg (6) has a pressure drop term - PDLEG(6), then PDLEG(6) will be multiplied by the conductance for leg (6) which is G(6), making the resulting term a flow. Thus, all external flows have no multiplication factor.

With both CALC1 and CALC2 filled, the TGAUSS subroutine is called to solve for pressures in the system. The answers are returned through the CALC1 array and then put into the PN array which contains all the system node pressures. Now a new flow is calculated for each leg in the system based on the recent calculation of the pressures. The new flow is equal to the difference of pressures between the nodes of the leg plus any constant pressure drops all multiplied by the conductance of the leg.

The solution for flows in all the legs are final when all the previous flows (Q) and the latest calculated flows (FLOW) are within a specified tolerance. For all flows if

$$\text{ABS}\,\frac{(\text{FLOW}-\text{Q(IT)})}{\text{ABS(FLOW)}+1} \leq .001 \qquad (1)$$

then the flows have converted.

If equation (1) is not satisfied in each leg of the system a new value of flow will be computed in each leg by the following equation:

$$Q(IT) = \frac{Q(IT) + FLOW}{2} \qquad (2)$$

These new flows will then be given to TLEGCAL for computation of new conductance values for another iteration. If all the legs do not converge after fifty iterations, the cycle will stop and all the current values will be used as the steady-state variables. Before transfer is made back to THYTR a last call is made to TLEGCAL to distribute pressure drops and flows for the steady state conditions.

Whenever the TPUMP51 subroutine is included in the simulation, a check is made to see if the pump outlet flow and pressure are within the operating characteristics. A typical pump outlet pressure/flow curve is shown in Figure 4.2-5.

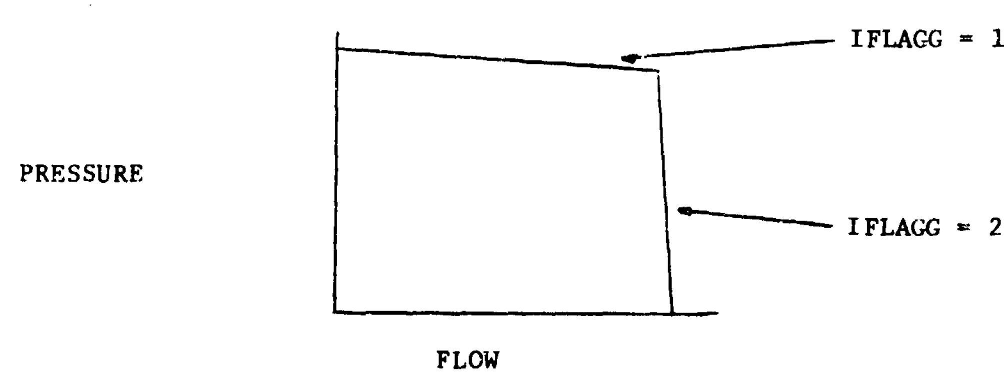


FIGURE 4.2-5

TYPICAL PRESSURE COMPENSATED VARIABLE DISPLACEMENT HYDRAULIC PUMP CHARACTERISTICS

If a balance occurs on LEG2 (IFLAGG=2), the pump outlet flow is checked. When it is below the rated flow the iteration is restarted with the pump model using the LEG 1 (IFLAGG=1) characteristic.

A check is also made to determine if the pump outlet flow is above rated flow and the iteration balanced on the proper pump leg. If not the iteration is restarted.

4.2.3 Computations. The only direct computation made in the solution of the steady state values in TCALC is the calculation of FLOW. The purpose of this is to establish an error tolerance in flows that is reduced through iterations to meet the convergence criteria as discussed in the previous section. The majority of the TCALC subroutine handles the bookkeeping necessary to manipulate the leg and node numbers to compute system pressures and flows.

4.2.4 Approximations. The coefficients of the CALC1 array are linearily approximated to represent the system conductances. Inherent approximations exist in some of the constant data in CALC2.

4.2.5 Limitations. Most limitations exist in the areas of physical discontinuities. TCALC was written to solve a flow balance in a system. Any flow discontinuities that occur, such as in a simple unbalanced actuator, must have mathematical formula to describe what happens to the flow. TCALC also requires the leg pressure drops to be continuous over a specified flow range. When this does not occur, as in a check valve, the proper input from the check valve subroutine must be fed to TCALC so it may respond to the changed conditions. Refer to Appendix A SSFAN Technical Manual (AFAPL-TR-76-43, Vol. VI) for a more thorough discussion on the limitations of TCALC.

4.2.6 Variable Names

Variables	Description	Dimensions
CALC1( )	Array of conductances	--
CALC2( )	Array of constants	--
FLOW	Latest value of leg flow	CIS
I	DO loop counter	--
ICOUNT	# of complete balances per time step	--
IFLAGG	Indicator for pump subroutine	--
G	Array of conductances	CIS/PSI
IFAIL,IFLAG	Indicators	--
IL,IM	Dummy variables	--
INEG( ),JNEG( )	Arrays containing location of off diagonal conductance values	--
JPCOL( )	Computational array	--
ITER	Iteration counter	--
IU,IV,J,JJ,JL, JX,JY,K,KOUNT, K1,2,LM,L1	Dummy variables	--
M	Number of nodes	--
ML	Total number of legs	--
PN( )	Array of node pressures	PSI
PDLEG( )	Location of pressure drops or increases	PSI
PEX	Array of external pressure constants	PSI
QL( )	Array of leg flows	CIS
QN( )	Flow gain or loss at a pressure node changed to an M matrix of constants	CIS

4.2.7 Subroutine Listing

```
      SUBROUTINE TCALC
**** REVISED JULY 07,1976 ****
      DOUBLE PRECISION CALC1.CALC2
      COMMON G(90),CALC2(55),JPCOL(55,20),CALC1(55,20)
      COMMON/ICC/ICOL(55,20),JRENT(55),JCENT(55)
      COMMON /STEADY/PN(90),QN(90),PEX(90),PDLEG(90),QL(90),
     + QA,QS,Q1,PUP,PDOWN,M,ML,NCPN,TERM,
     +  LEGN,ICON,INV,INX,INZ,NUP(90),NDWN(90),NELEM(90),
     +ILEGAD(90),ILEG(1000),IFLAGG,CON2,QQQ,ITER,ICOUNT
      DIMENSION IDIAG(55),JCOL(55,55),JNEG(90),INEG(90),ID(20)
      INTEGER TEST2,TEST,CON2
      EQUIVALENCE(JPCOL(1,1),JCOL(1,1))
      IF(NCPN.EQ.0)WRITE(6,900)
  900 FORMAT(1H1,50X,30HSTEADY STATE CALCULATION DATA )
      DO 5 I=1,M
      QN(I)=0.0
      PEX(I)=0.0
 5    PN(I)=0.0
      DO 6 I=1,ML
    6 PDLEG(I)=0.0
      DO 11 I=1,55
      DO 11 J=1,55
 11   JCOL(I,J)=0
C-----BUILD JCOL IN COMPRESSED FORM
      DO 10 K=1,ML
      I=NUP(K)
      J=NDWN(K)
      DO 10 K1=1,2
      J1=I
      IF (K1.EQ.2)J1=J
      DO 10 K2=1,2
      J2=I
      IF (K2.EQ.2)J2=J
      DO 10 K3=1,20
      J3=JCOL(J1,K3)
      IF (J3.NE.0)GO TO 10
      JCOL(J1,K3)=J2
      J2=0
 10   IF (J3.EQ.J2)J2=0
C-----BUILD JRENT,JCENT:REORDER JCOL:BUILD IDIAG
      DO 15 K=1,M
      KOUNT=0
      DO 20 K1=1,M
      DO 20 K2=1,20
 20   IF (JCOL(K1,K2).NE.0)KOUNT=KOUNT+1
      JCENT(K)=KOUNT
      KOUNT=0
      DO 25 K3=1,20
 25   IF (JCOL(K,K3).NE.0)KOUNT=KOUNT+1
      JRENT(K)=KOUNT
```

4.2.7 (Continued)

```
      DO 30 K4=1,KOUNT
   30 ID(K4)=JCOL(K,K4)
      DO 35 K5=1,KOUNT
      TEST=0
      DO 40 K6=1,KOUNT
      IF (ID(K6).LT.TEST)GO TO 40
      TEST=ID(K6)
      TEST2=K6
   40 CONTINUE
      K7=KOUNT+1-K5
      JCOL(K,K7)=TEST
   35 ID(TEST2)=0
      K8=JRENT(K)
      DO 15 KOUNT=1,K8
   15 IF (JCOL(K,KOUNT).EQ.K)IDIAG(K)=KOUNT
C-----BUILD INEG,JNEG
      DO 45 K=1,ML
      I=NUP(K)
      J=NDWN(K)
      I1=JRENT(I)
      J1=JRENT(J)
      DO 50 KOUNT=1,I1
   50 IF (JCOL(I,KOUNT).EQ.J)JNEG(K)=KOUNT
      DO 45 KOUNT=1,J1
   45 IF (JCOL(J,KOUNT).EQ.I)INEG(K)=KOUNT
C     INITIALIZE CALC1 AND CALC2 ARRAYS TO ZERO
      CON2=-1
      IFLAGG=1
      ENTRY TSCALC
      ICOUNT=0
   80 ICOUNT=ICOUNT+1
      ITER=1
      IF(NCPN.EQ.0.OR.NCPN.EQ.2)WRITE (6,9004)ICOUNT
 9004 FORMAT(/,11H   ICOUNT =,I5)
      IF(NCPN.EQ.0.OR.NCPN.EQ.2)WRITE(6,910)
  910 FORMAT(//,24X,10HFLOW QUESS,4X,13HPRESSURE DROP,6X,9HLEG DELTP,
     + 9X,3HPUP,12X,5HPDOWN,9X,11HCONDUCTANCE,/)
  200 CONTINUE
      IF(NCPN.EQ.0)WRITE (6,9003)ITER,IFLAGG
 9003 FORMAT (/,20H  ITERATION NUMBER =,I10,10X,10H  IFLAGG =,I10)
      DO 220 L1=1,M
      DO 210 K1=1,20
      ICOL(L1,K1)=0
  210 CALC1(L1,K1)=0.
      PEX(L1)=0.0
      QN(L1)=0.0
  220 CALC2(L1)=0.
      DO 221 L1=1,ML
  221 PDLEG(L1)=0.0
C     COMPUTE G*S FOR CALC ARRAYS
      CALL TLEGCAL
```

```
      DO 9099 K=1,ML
      I=NUP(K)
      J=NDWN(K)
      L=IDIAG(I)
      LM=IDIAG(J)
      CALC1(I,L)=CALC1(I,L)+G(K)
      CALC1(J,LM)=CALC1(J,LM)+G(K)
      L=JNEG(K)
      LM=INEG(K)
      CALC1(I,L)=CALC1(I,L)-G(K)
      CALC1(J,LM)=CALC1(J,LM)-G(K)
 9099 CONTINUE
      DO 700 IL=1,55
      DO 700 JL=1,20
 700  ICOL(IL,JL)=JPCOL(IL,JL)
      DO 400 JX=1,ML
      IF(PDLEG(JX).EQ.0.)GO TO 400
      JY=NUP(JX)
      CALC2(JY)=CALC2(JY)-PDLEG(JX)*G(JX)
      JY=NDWN(JX)
      CALC2(JY)=CALC2(JY)+PDLEG(JX)*G(JX)
  400 CONTINUE
      DO 60 I=1,M
      J=IDIAG(I)
      CALC2(I)=CALC2(I)+QN(I)
 60   CALC1(I,J)=CALC1(I,J)+PEX(I)
C     WRITE(6,2005)((CALC1(I,J),J=1,20),I=1,M)
C     WRITE(6,2005)(CALC2(I),I=1,M)
C     WRITE(6,2004)((ICOL(I,J),J=1,20),I=1,M)
 2004 FORMAT(1X,20I6)
 2005 FORMAT(1X,10E12.5)
C     WRITE(6,2005)(PEX(I),I=1,M)
C     WRITE(6,2005)(PDLEG(I),I=1,ML)
C     WRITE(6,2005)(QN(I),I=1,M)
      CALL TGAUSS(M,ITER)
      DO 410 IM=1,M
 410  PN(IM)=CALC1(IM,1)
      IF(NCPN.EQ.0)WRITE(6,9000)(PN(I),I=1,M)
      IF(NCPN.EQ.0)WRITE(6,9001)
 9000 FORMAT(1H0,(5X,14HNODE PRESSURES,2X,8F12.3,/))
 9001 FORMAT(1H0)
      IFAIL=0
C     CALCULATE NEW FLOW RATES
      DO 435 IT=1,ML
      IU=NUP(IT)
      IV=NDWN(IT)
      QOLD=QL(IT)
      FLOW=((PN(IU)+PDLEG(IT)-PN(IV))*G(IT))
C     TEST NEW FLOW RATES
      IF(ABS(FLOW-QOLD)/(ABS(FLOW)+1.).GT.0.001)GO TO 436
```

4.2.7 (Continued)

```
C     RECALCULATE FLOW RATES
      QL(IT)=FLOW
      GO TO 435
  436 QL(IT)=.5*QOLD+.5*FLOW
      IFAIL=1
  435 CONTINUE
      IF(IFAIL.EQ.0)GO TO 520
      IF(ITER.EQ.50)WRITE(6,999)
  999 FORMAT(10X,44H**** WARNING EXCEEDED 50 ITERATIONS IN TCALC,
     + 13H-PROGRAM STOP,//)
      IF(ITER.EQ.50)STOP
      ITER=ITER+1
      GO TO 200
  520 CONTINUE
C     MAKE A LAST CALL TO ALL LEGS TO DISTRIBUTE PRESSURE
C     DROPS AND FLOWS CALCULATED FOR STEADY STATE CONDITIONS
      DO 521 I=1,ML
  521 PDLEG(I)=0.0
      CALL TLEGCAL
      IF(CON2.EQ.-1)GO TO 522
C     CHECK PUMP CALCULATION
      IF (ICOUNT .EQ. 2)GO TO 522
      IF(IFLAGG.EQ.2)GO TO 523
      IF(QL(CON2).LE.QQQ)GO TO 522
      IFLAGG=2
      GO TO 80
  523 IF(QL(CON2).GT.QQQ)GO TO 522
      IFLAGG=1
      GO TO 80
  522 IF(NCPN.EQ.2)WRITE(6,9000)(PN(I),I=1,M)
      IF(NCPN.EQ.2)WRITE(6,9001)
      IF(NCPN.EQ.2)WRITE(6,9002)(QL(I),I=1,ML)
 9002 FORMAT(1H0,(5X,9HLEG FLOWS,7X,8F12.3,/))
      IF(NCPN.EQ.2)WRITE(6,9001)
      RETURN
      END
```

4.3 SUBROUTINE TLEGCAL

TLEGCAL is called by the TCALC to obtain a leg conductance and fixed pressure drops for a given flow guess for all the system.

The constant pressure drop such as that across a check valve, actuator, or pump, is passed via TERM or PDLEG(INLEG) in COMMON/STEADY. A positive PDLEG(NLEG) is a pressure rise such as at a pump, a negative is a drop such as across a check valve. The leg conductance is passed via G(NLEG) in common.

TLEGCAL obtains the line and component pressure drops by calling all the elements in the leg. The leg conductance is computed by dividing the leg flow by the leg pressure drop.

A flowchart of TLEGCAL's organization is shown in Figure 4.3-1.

4.3.1 Theory

LEGCAL calls the elements in a leg to determine the pressure drop for a given flow.

The leg conductance (inverse of resistance) is calculated from the leg pressure drop, including the constant pressure drop value

G(NLEG) = QA/ABS(DELTP)

where

DELTP = PN(NUP(NLEG))-PUP+TERM

The conductance is always positive. Using this formula the conductance value is flow dependent. It has to be updated whenever the flow guess is changed.

4.3.2 Assumptions

The assumption that the pressure drop can be described using the leg pressure drop is generally valid. If for some reason an element in a leg cannot be described in this manner, then a pseudo description can

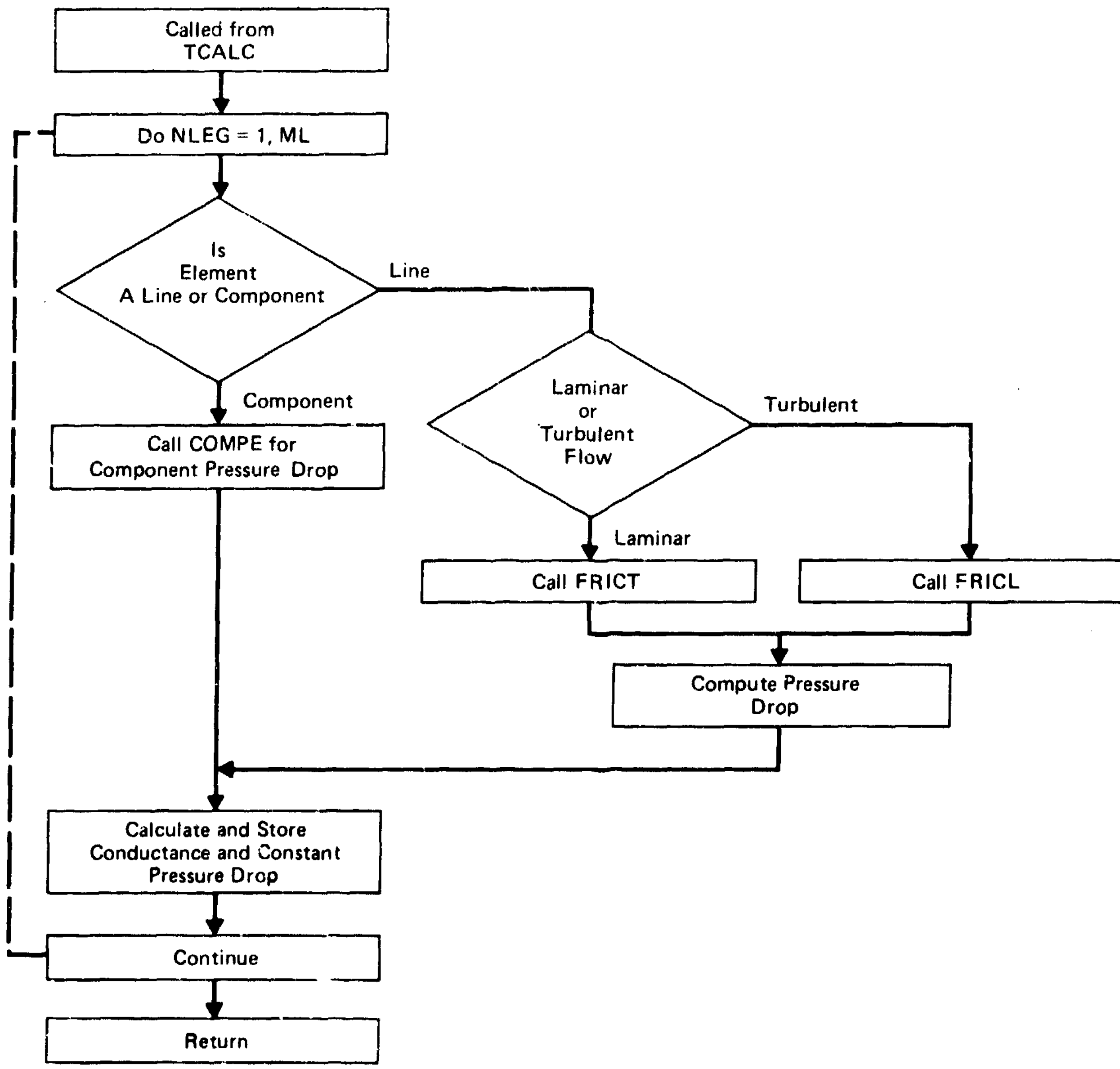


FIGURE 4.3-1
TLEGCAL ORGANIZATION

be used without loss of accuracy. This could involve generating a formula of the form

$$\Delta P = K_1 + K_2 Q + K_3 Q^{1.75} + K_4 Q^2$$

where Q = leg flow

The line and component subroutines would then provide the values to the K_1, K_2, K_3 and K_4 constants.

4.3.3 Computation Method

The variable Q1, the new flow guess, is first split into its absolute value and its sign, ± 1.0. The up and downstream node pressures for the leg are taken from the PN( ) array.

Each element in a leg is called and the pressure drop through the line or component is calculated and subtracted from the upstream pressure, PUP. Once the entire leg pressure drop has been determined the new conductance value is computed. The variables IND, and KNEL are the component number and the connection number respectively.

The common variables INZ and INX are set equal to the number of elements in the leg and the actual element that is being calculated respectively. This allows particular component subroutines to determine which end of the leg they are connected to, and hence which node is located at the component.

4.3.4 Approximations

The use of a formula requires some approximations but these are usually related to approximations in the component model and are an integral part of the component model. In general this method is good but it could be easily extended to a higher order approximation if it was found desirable.

The pressure drop for a line is computed based on the average temperature and pressure in the line. The average value is determined from the values at the upstream and downstream line locations. Line pressure losses due to bends are calculated using an energy bend loss coefficient, which is computed in the TLINEA subroutine.

4.3.5 Limitations

So far we have not found any limitations to the technique used in TLEGCAL itself.

However, some of the component subroutines called by TLEGCAL such as the bootstrap reservoir, pump and actuators, are complicated by the interaction between the flow guesses, flow direction and node pressures.

Some of these subroutines use calculations which, though conforming to the basic calculation technique, do not fall into any simple category and have to be treated individually.

4.3.6 Variable Names

Variable	Description	Dimensions
DELTP	Line Pressure Drop	PSI
INZ	Number of Elements in Leg	--
I	Address of Leg Data in ILEG	--
K	Ith Element in a Leg	--
KNEL	Component Connection Number	--
IND	Component or Line Number	--
ML	Total Number of Legs	--

Variable	Description	Dimensions
PAVE	Average Line Pressure	PSI
PUP	Flow Dependent Leg Pressure Drop	PSI
QA	ABS Value of Leg Flow	CIS
QT	Leg Transition Flow	CIS
Q1	Leg Flow Guess	--
QS	Flow Sign CIS	--
TAVE	Average Line Temperature	°F

For variables in common refer to Paragraph 3.3.

4.3.7 Subroutine Listing

```
      SUBROUTINE TLEGCAL
C**** REVISED AUGUST 5, 1976 ****
      COMMON G(90)
      COMMON /TRANS/P(300),Q(300),C(300),TC(300),TW(300),TF(300),
     + ACF(300),ACW(300),DXF(300),TIME,DELT,PI,NLINE,NEL
      COMMON /COMP/LTYPE(99),NC(99),KTEMP(99),IND,IENTR,INEL
      COMMON /LINE/PARM(150,4),TLW(2000),TLF(2000),LSTART(150),
     + NLSEG(150)
      COMMON /STEADY/PN(90),QN(90),PEX(90),PDLEG(90),QL(90),
     + QA,QS,Q1,PUP,PDWN,M,ML,NCPN,TERM,
     +  LEGN,ICON,INV,INX,INZ,NUP(90),NDWN(90),NELEM(90),
     +ILEGAD(90),ILEG(1000),IFLAGG,CON2,QQQ
C     FIND THE SIGN OF THE FLOW GUESS AND ITS ABSOLUTE VALUE
      DO 200 NLEG=1,ML
 210  TERM=0.0
      INV=1
      Q1=QL(NLEG)
      QA=ABS(Q1)
      IF(QA.LE..00001)QA=.00001
      QS=SIGN(1.0,Q1)
C     CALCULATE THE FORMULAE FOR THE LEG PRESSURE DROP
      INEL=NLEG
      INZ=NELEM(NLEG)
C     INZ - NO OF ELEMENTS IN A LEG
C     INEL - LEG NUMBER
      PUP=PN(NUP(NLEG))
      PDWN=PN(NDWN(NLEG))
C     WRITE(6,900)NLEG,INZ,NUP(NLEG),NDWN(NLEG)
  900 FORMAT(10X,5I10)
C     WRITE(6,910)PUP,PDWN,Q1
  910 FORMAT(10X,5E12.5)
C     CALL EACH ELEMENT IN THE LEG
C
      I=ILEGAD(NLEG)
      DO 600 K=1,INZ
C     INX - CURRENT ELEMENT NO. IN LEG
      INX=K
      IND=ILEG(I)
      KNEL=ILEG(I+1)
      I=I+2
      ICON=KNEL
C      ICON - CONNECTION NO.
C    IF THE ELEMENT IS A LINE GO TO 500
      IF(IND.EQ.0) GO TO 500
      CALL COMPE
      GO TO 600
```

4.3.7 (Continued)

```
C *** THIS SECTION ADDS THE VALUES INTO THE FORMULAE FOR THE LINES
C
 500  CONTINUE
      LOC=KNEL*2-1
      LOCD=KNEL*2
C     WRITE(6,503) LOC,TF(LOC),PUP
C 503 FORMAT(3X,I110,2E12.5)
      TAVE=(TF(LOC)+TF(LOCD))/2
      PAVE= PUP
      QT=PARM(KNEL,4)*VISC(TAVE,PAVE)
      IF(QA.GT.QT)GO TO 505
      DELTP=QA*FRICL(KNEL,TAVE,PAVE)
      GO TO 598
 505  DELTP=FRICT(KNEL,TAVE,PAVE)*QA**1.75
  598 DELTP=DELTP+RHO(TAVE,PAVE)*PARM(KNEL,3)*QA**2
      P(LOC)=PUP
      IF(TERM.GT.0.0.AND.INV.EQ.1)P(LOC)=TERM
      PUP=P(LOC)-DELTP
      P(LOCD)=PUP
      Q(LOC)=-Q1
      Q(LOCD)=Q1
  600 CONTINUE
      DELTP=PN(NUP(NLEG))-PUP+TERM
      IF(DELTP.EQ.0.0)DELTP=.0001
      G(NLEG)=QA/ABS(DELTP)
      IF(NCPN.EQ.0)WRITE(6,50)NLEG,Q1,PDLEG(NLEG),DELTP,
     + PN(NUP(NLEG)),PN(NDWN(NLEG)),G(NLEG)
      IF(INV.EQ.0)GO TO 200
      PDLEG(NLEG)=0.0
   50 FORMAT(13H      LEG NO ,I3,5F16.5,E20.8)
 200  CONTINUE
      RETURN
      END
```

APPENDIX U
LINE DATA
HYTTHA USER MANUAL (AFAPL-TR-76-43, VOL. VII)

5.0 LINE DATA

The number of cards used in this group will be twice the number of lines entered on card 2 (two cards for each line). An error message will be written when the number of lines exceeds the maximum number specified in block data and the program will stop. A line number may not be omitted or used twice.

To differentiate between rigid lines and flexible hoses the material type will be used. The same mathematical equations are involved with both types of lines so the same routine is used for each.

A line temperature indicator is entered in column 15. When a zero is entered the line temperature values specified on the second line data card will be used. A one in column 15 will select the default values in the main program.

The environment indicator is entered in columns 16-20. A zero value defaults to the atmospheric temperature on card two. A numerical value must correspond to the appropriate environment number.

Columns 24-25 contain the number of bends in the line. Eleven is the maximum number. The actual bend angles are entered on the remainder of the card in integer I5 format.

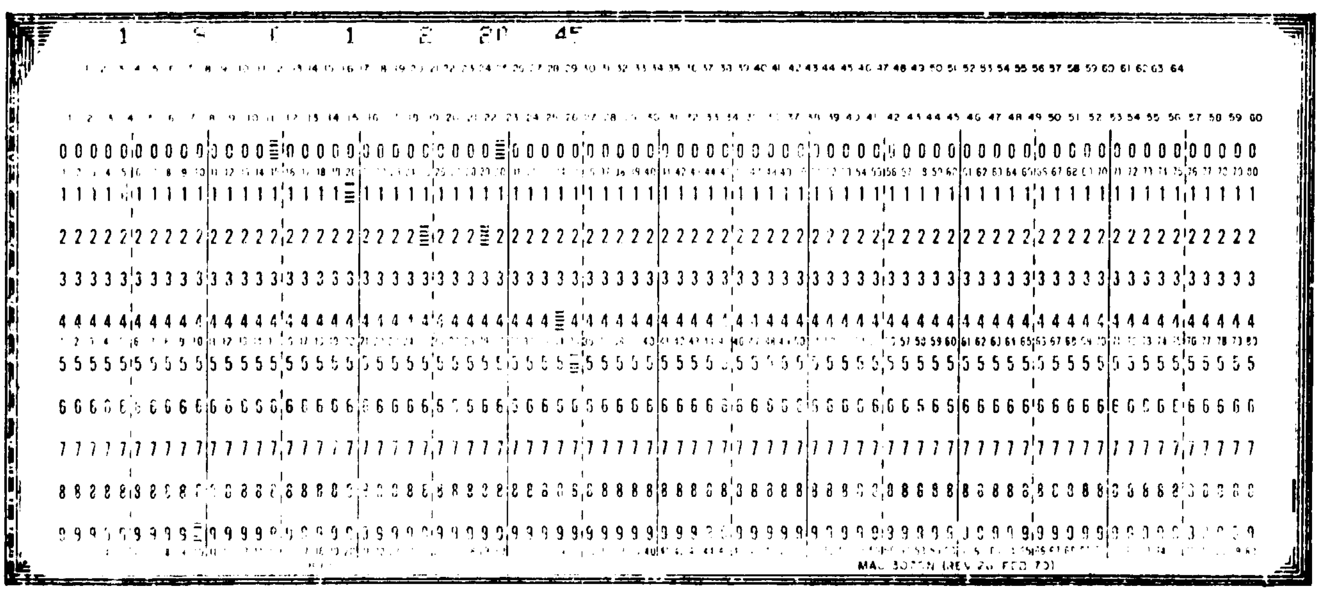
Card number two of the line data is self explanatory except for the heat transfer coefficient. If not input by user then the program will set it equal to 0.0069 which is a value nearly equal to that for still air.

TLINEA

CARD NUMBER 1

COLUMN	FORMAT	DATA	Dimensions
1-5	I5	Line Number	--
6-10	I5	Material Type	--
11-15	I5	Line Temperature Indicator	--
16-20	I5	Environment Indicator	--
21-25	I5	Number of Bends (11 Maximum)	--
26-30	I5	First Bend Angle	Degrees
31-35	I5	(Continue entering bend angles until	
36-40	I5	the number specied in columns 24-25 is	
41-45	I5	reached).	
46-50	I5		
51-55	I5		
56-60	I5		
61-65	I5		
66-70	I5		
71-75	I5		
76-80	I5		

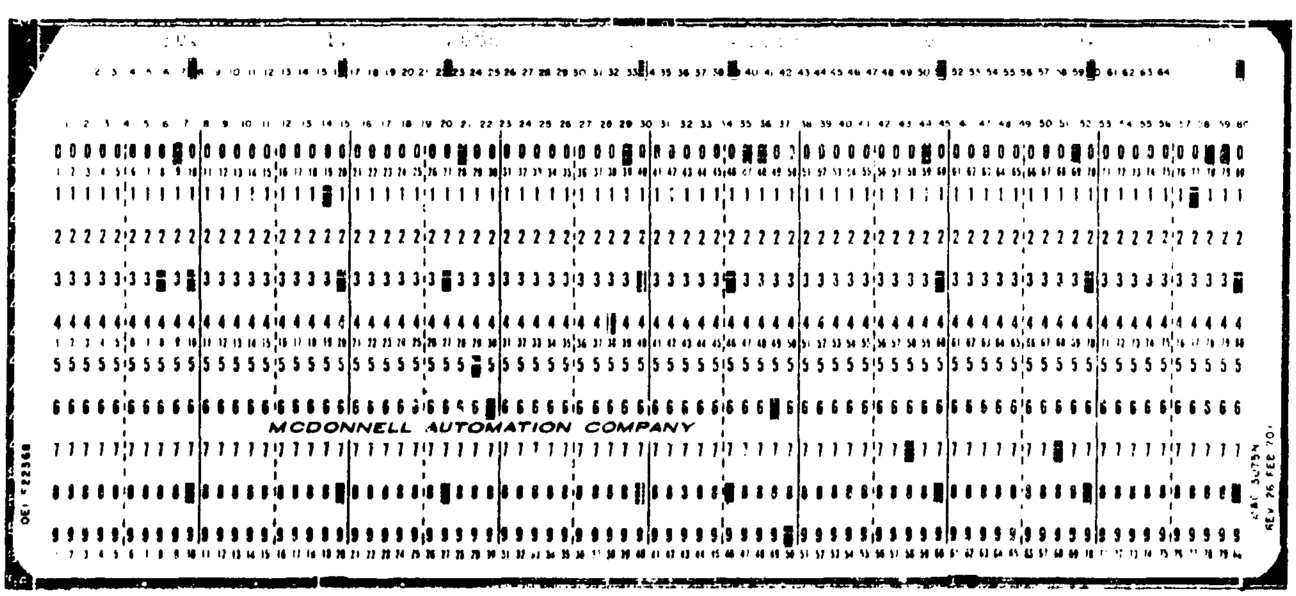
EXAMPLE CARD



CARD NUMBER 2

COLUMN	FORMAT	DATA	DIMENSIONS
1-10	E10.0	TOTAL LINE LENGTH INCLUDING FITTINGS	IN.
11-20	E10.0	OUTSIDE LINE DIAMETER	IN.
21-30	E10.0	LINE WALL THICKNESS	IN.
31-40	E10.0	LENGTH OF LINE SEGMENT (DELTAX)	IN.
41-50	E10.0	HEAT TRANSFER COEFFICIENT, WALL TO ATMOSPHERE	WATT/IN^2-°F
51-60	E10.0	SURROUNDING STRUCTURE TEMPERATURE	°F
61-70	E10.0	SURROUNDING ATMOSPHERIC TEMPERATURE	°F
71-80	E10.0	HYDRAULIC FLUID INITIAL TEMPERATURE	°F

EXAMPLE CARD



The energy loss coefficient for bends in a tube or hose is calculated in the 1000 section. The bend ratio (BRAT) for a tube is assumed to be

3.* (LINE O.D./LINE I.D.)

The bend ratio is 8. for hoses.

The DO loop parameter is set to the number of bends in the tube (NBEND), which is read from the line data card. The bend in the tube being read is stored in the NBAN( ) array. As each bend angle is read from NBRAN( ), the energy loss coefficient is calculated and summed into a temporary storage location (ECOEF).

The energy loss coefficient is proportional to the pressure drop through the bend divided by the dynamic pressure or

$$f = \frac{\Delta P}{\frac{\rho}{2} v^2} \qquad ( )$$

Where

f = Bend energy loss coefficient

ΔP = Pressure Drop (psid)

ρ = Fluid Density ($\text{lb-sec}^2/\text{in}^4$)

v = Fluid Velocity (in/sec)

Modifying equation ( ) to fluid flowrate and solving for pressure drop yields

$$\Delta P = \frac{1}{A^2} \; f \; \frac{\rho}{2} \quad Q$$

Where A = Line cross sectional area (in^2)

When all the bend angles on one line are processed, the bend energy loss coefficient is multiplied by (0.5 / A^2) and stored in PARM(N,3). The TLEGCAL subroutine multiplies this term by the fluid density and current flowrate for the appropriate tube. Further discussion and the derivation of the bend energy loss coefficient is found in Appendix C of AFAPL-TR-76-43 Volume VI.

SUBROUTINE LISTING

```
      SUBROUTINE TLINEA
C**** REVISED DECEMBER 1979 ****
      COMMON /TRANS/P(300),Q(300),C(300),TC(300),TW(300),TF(300),
     + ACF(300),ACW(300),DXF(300),TIME,DELT,PI,NLINE,NEL
      COMMON /LIMIT/MNLINE,MNEL,MNLEG,MNNODE,MNPLOT,MNLPTS,MDS
      COMMON /LINE/PARM(150,4),TLW(2000),TLF(2000),LSTART(150),
     + NLSEG(150)
      COMMON /COMP/LTYPE(99),NC(99),KTEMP(99),IND,IENTR,INEL
      COMMON /FLUID/ATPRES,CF,CPFN,FTEMP,PROP(13,3),IDAT,AT,IDSTT,STT,
     +IDWT,WT,IDTEMP,IDFLT,FLT
      COMMON/ENVIRON/NENV,VAT(5),VSTT(5),VWT(5)
      DIMENSION TA(300),TST(300),UAW(300),TFO(300),DELTAX(150),
     +TCO(300),SPMAF(150),SPMAW(150),PLENGTH(150)
      DIMENSION LC(300),ASAW(150),ASFW(150),FNM(150),CW(150)
     + ,WNM(150),FWTEMP(300),NBAN(11),A(2,2),B(2)
      EQUIVALENCE (C(1),LC(1)),(PARM(1,1),PLENGTH(1))
      DATA SIGMA/.349E-11/,SHAPF/.96/,EPSION/.3/
      REN=1200.
      IF(IENTR)1000,2000,3000
 1000 CONTINUE
C
C     INO     =INDIVIDUAL LINE NUMBER
C     LINET=LINE TYPE
C     LINETH(N)=LINE PLENGTH
C     DIA  =OUTSIDE DIAMETER
C     WTHICK=WALL THICKNESS
C     MTYPE =MATERIAL TYPE
C     DELTAX(INO)=DISTANCE OF SEGMENTS
C     UAW(N)=HEAT TRANSFER COEFF. AMBIENT TO WALL
C     TA(N) =TEMP  OF AMBIENT, DEG. F
C     TLF(N) =TEMP. OF FLUID, DEG. F
C     TST(N)=TEMP. OF STRUCTURE, DEG. R
C
C     PARM(N,1) = LINE LENGTH
C     PARM(N,2) = INSIDE LINE DIA-DIAINS
C     PARM(N,3) = BEND ENERGY LOSS COEFFICIENT
C     PARM(N,4) = TRANSISTION FLOW
      LSTART(1)=1
      WRITE(6,400)
  400 FORMAT(//11H  LINE DATA,/10H  LINE NO.,5X,6HLENGTH,5X,
     + 8HINTERNAL,7X,4HWALL,8X,6HDELTAX,8X,7HAMBIENT,4X,9HSTRUCTURE,
     + 6X,5HFLUID,7X,8HMATERIAL,5X,6HLDTEMP,2X,7HENVTYPE,/29X,3HDIA,7X,
     + 9HTHICKNESS,21X,4HTEMP,8X,4HTEMP,9X,4HTEMP,10X,4HTYPE)
      DO 1100 INO=1,NLINE
      READ(5,433) N,MTYPE,LDTEMP,IENV,NBEND,(NBAN(I),I=1,NBEND)
  433 FORMAT(16I5)
      READ(5,439) PLENGTH(INO),DIA WTHICK,DELTAX(INO),UAW(INO),
     + TST(INO) TA(INO),FLTEMP
  439 FORMAT(8E10.0)
      IF(INO.NE.N) WRITE(6,430) N
```

LISTING (Continued)

```
  430 FORMAT(1X,43H THE LINE CARDS ARE OUT OF ORDER AT NUMBER ,I5)
      IF(IDTEMP .EQ. 0 .OR. LDTEMP .EQ. 0) GOTO 440
      IF( IDAT .EQ. 1)TA(INO)=AT
      IF( IDSTT .EQ. 1)TST(INO)=STT
      IF( IDFLT .EQ. 1)FLTEMP=FLT
  440 IF(IENV.EQ.0)GOTO 450
C ***  SET DATA TO PRORER ENVIRONMENT ***
      TA(INO)=VAT(IENV)
      TST(INO)=VSTT(IENV)
C
  450 M=N*2
      LC(M)=1
      LC(M-1)=1
      IF(LINET.LT.10) GO TO 65
      LC(M)=-1
      LINET=LINET-10
   65 CONTINUE
C     CALCULATE NUMBER OF SEGMENTS
      IF(UAW(INO).EQ.0.0) UAW(INO)=0.0069
      IF(DELTAX(INO).EQ.0.0) DELTAX(INO)=36.
      IF(FLTEMP.EQ.0.0) FLTEMP=FTEMP
      NLSEG(INO)=PLENGTH(N)/DELTAX(INO)
      LSTART(INO+1)=LSTART(INO)+NLSEG(INO)
      IF(LSTART(INO+1).GT.MNLPTS)GO TO 252
      IND=INO
      INAD=INO*2
      INAU=INO*2-1
      RHOW=PROP(MTYPE,2)
      CPWN=PROP(MTYPE,1)
      FWTEMP(INAD)=FLTEMP
      FWTEMP(INAU)=FLTEMP
      DIAINS=DIA-2.0*WTHICK
C     COMPUTE ENERGY LOSS COEFFICIENT FOR BENDS
      ECOEF=0.0
      IF(NBEND.EQ.0)GO TO 550
      BRAT=3.*(DIA/DIAINS)
      IF(MTYPE.EQ.13)BRAT=8.
      DO 500 I=1,NBEND
      NBA=NBAN(I)
      IF(NBA.GT.180)GO TO 510
      A1=-3.97626E-10*NBA**4+2.8475E-7*NBA**3-9.23298E-5*NBA**2
      A1=A1+1.74517E-2*NBA
      GO TO 520
  510 A1=1.39+(NBA-180)*3.3333E-3
  520 B1=.206982*BRAT**(-.49421)
      IF(BRAT.GT.30.)B1=.0385411-(BRAT-30.)*4.E-4
      ECFF=A1*B1
      ECOEF=ECOEF+ECFF
  500 CONTINUE
```

```
  550 CW(INO)=PROP(MTYPE,3)
      ACW(INAD)=PI*(DIA**2-DIAINS**2)/4 0
      ACW(INAU)=ACW(INAD)
      WNM(IND)=ACW(INAD)*RHOW*DELTAX(INO)
      SPMAW(IND)=WNM(IND)*CPWN
      ASAW(IND)=PI*DIA*DELTAX(INO)
      ACF(INAD)=PI*DIAINS**2/4.0
      ACF(INAU)=ACF(INAD)
      ASFW(IND)=PI*DIAINS*DELTAX(INO)
      TF(INAU)=FLTEMP
      TF(INAD)=FLTEMP
      TW(INAU)=FLTEMP
      TW(INAD)=FLTEMP
      PARM(N,2) = DIAINS
      PARM(N,3) = ECOEF*.5/(ACF(INAU)*ACF(INAU))
      PARM(N,4) = .7854*REN*DIAINS
      WRITE(6,410)N PLENGTH(N),DIAINS,WTHICK,DELTAX(N),TA(N),
     + TST(N),FLTEMP,MTYPE,LDTEMP,IENV
  410 FORMAT(/1X,I5,8X,F8.4,4X,F8.4 5X,F8.4,5X,F8.4,7X,F8.4 4X,F8.4
     + ,5X,F8.4,6X,I4,6X,I5,7X,I2)
      TST(IND)=(TST(IND)+460.)**4
 1100 CONTINUE
      IF(NLINE.GT.MNLINE-1) GO TO 252
      RETURN
  252 WRITE(6,475) NLINE,LSTART(NLINE),NLSEG(NLINE)
  475 FORMAT(5X,25HERROR IN SUBROUTINE TLINE,3I10)
      STOP 5101
 2000 CONTINUE
C     INITALIZING ALL TEMP. IN LINE
      DO 2012 I=1,NLINE
      XDEL=ABS(Q(I*2-1))*DELT/(ACF(I*2-1))
      IF(XDEL.LE.DELTAX(I))GO TO 2015
      DELTAX(I)=XDEL
      IF(DELTAX(I).GT.PLENGTH(I))DELTAX(I)=PLENGTH(I)
      WRITE(6,900)I,DELTAX(I)
  900 FORMAT(10X,18HTHE DELTAX IN LINE,I5,22H HAS BEEN CORRECTED TO,
     + F15.5,7H INCHES,//)
 2015 NLSEG(I)=PLENGTH(I)/DELTAX(I)
      LSTART(I+1)=LSTART(I)+NLSEG(I)
      ASFW(I)=PI*PARM(I,2)*DELTAX(I)
      WNM(I)=ACW(I*2)*RHOW*DELTAX(I)
      SPMAW(I)=WNM(I)*CPWN
 2012 ASAW(I)=PI*PARM(I,2)*DELTAX(I)
 2013 INO=1
```

```
 2010 J=LSTART(INO)
      INAD=INO*2
      INAU=INO*2-1
      AAA=ACF(INAU)
      DDD=PARM(INO,2)
      UFWIL=UFW(AAA,DDD,ABS(Q(INAU)),TF(INAU),P(INAU))
      TLW(J)=FWTEMP(INAU)
 2020 TLF(J)=FWTEMP(INAU)
      JM1=J-1
      IF(JM1.EQ.0) JM1=1
      TLW(J)=TLF(J)
C     TLW(J)=(UAW(INO)*ASAW(INO)*TA(INO)-UFWIL*ASFW(INO)*TLF(J)
C    1 +SIGMA*EPSION*SHAPF*ASAW(INO)*(TST(INO)))/(UAW(INO)*ASAW
C    2 (INO)-UFWIL*ASFW(INO)+SIGMA*EPSION*SHAPF*ASAW(INO)*TLW(JM1)**3)
      J=J+1
      IF(J.LE.(LSTART(INO)+NLSEG(INO)-1)) GO TO 2020
      C(INAU)=CW(INO)
      C(INAD)=CW(INO)
      TFO(INAU)=TF(INAU)
      TFO(INAD)=TF(INAD)
      TCO(INAU)=TC(INAU)
      TCO(INAD)=TC(INAD)
      DXF(INAU)=0.5*DELTAX(INO)
      DXF(INAD)=DXF(INAU)
      LSTART(INO+1)=LSTART(INO)+NLSEG(INO)
      INO=INO+1
      IF (INO.LE.NLINE) GO TO 2010
      RETURN
 3000 CONTINUE
      DO 3550 INO=1,NLINE
      INAU=2*INO-1
      INAD=2*INO
      IL=LSTART(INO)
      AAA=ACF(INAU)
      DDD=PARM(INO,2)
      RHOIL=386.4*RHO(TLF(IL),P(INO))
      UFWIL=UFW(AAA,DDD,ABS(Q(INO)),TLF(IL),P(INO))
      FNM(INO)=AAA*RHOIL*DELTAX(INO)
      SPMAF(INO)=FNM(INO)*CPFN
      N=NLSEG(INO)
      M=LSTART(INO)
      JJ=M+N-1
C     JJ IS THE LAST NODE
      J=M
C     J IS THE FIRST NODE
      IS=1
      IL=INO*2-1
      IF(Q(IL).LT.0.0) GO TO 3100
      IL=IL+1
      IS=-IS
      J=JJ
C     J IS THE LAST NODE
```

LISTING (Continued)

```
      JJ=M
      M=J
 3100 CONTINUE
      C1=ABS(Q(IL))*RHOIL*CPFN
      C3=UAW(INO)*ASAW(INO)
      CIP=SIGMA*EPSION*SHAPF*ASAW(INO)
      B9=UFWIL*ASFW(INO)
      RMF=ABS(Q(IL))*RHOIL
C     BEGINNING CALCULATION OF LINE TEMPERATURES
      IF(Q(IL).LT.0.0) GO TO 3033
      CID1=TC(IL+IS)
      CID2=TF(IL+IS)
      TC(IL+IS)=TC(IL)
      TC(IL)=CID1
      TF(IL+IS)=TF(IL)
      TF(IL)=CID2
 3033 TFP=TF(IL)*2.0-TFO(IL)
      TCP=TC(IL)*2.0-TCO(IL)
      TCO(IL)=TC(IL)
      TFO(IL)=TF(IL)
      TCO(IL+IS)=TC(IL+IS)
      TFO(IL+IS)=TF(IL+IS)
      IF(N.GT.1)GO TO 3200
C     THERE IS ONLY ONE SEG.
 3050 R1=CF/(2.0*DXF(INAU)/ACF(INAU)+RMF*DELT/(ACF(INAU)
     +  **2*RHOIL))
      R3=(CW(INO))/(2.0*DXF(INAU)/ACW(INAU))
      R4=R3
      A2=SPMAW(INO)/DELT+R3+B9+C3+R4
      A9=SPMAF(INO)/DELT+C1+R1+B9
      A(1,1)=A9
      A(1,2)=-B9
      A(2,1)=-B9
      A(2,2)=A2
      B(1)=SPMAF(INO)*TLF(J)/DELT+(R1+C1)*TFP
      B(2)=SPMAW(INO)*TLW(J)/DELT+R3*TCP+R4*TC(IL+IS)
     + +C3*TA(INO)+CIP*TST(INO)-CIP*(TLW(J)+460.)**4
      CALL SIMULT(A,B,2,IERROR)
      C9=B(1)
      C12=B(2)
      TLF(J)=B(1)
      TLW(J)=B(2)
      GO TO 3500
 3200 CONTINUE
C     FIRST LINE SEG.
      R1=CF/(2.0*DXF(INAD)/ACF(INAD)+RMF*DELT/(ACF(INAD)
     + **2*RHOIL))
      R3=1.0/(2.0*DXF(INAU)/(ACW(INAU)*CW(INO)))
      R4=R3
      A9=SPMAF(INO)/DELT+R1+B9+C1
      A2=SPMAW(INO)/DELT+R3+R4+B9+C3
      A(1,1)=A9
      A(1,2)=-B9
      A(2,1)=-B9
```

```
      A(2,2)=A2
      B(1)=SPMAF(INO)*TLF(J)/DELT+(R1+C1)*TFP
      B(2)=SPMAW(INO)*TLW(J)/DELT+R3*TCP+R4*TLW(J+IS)
     ++C3*TA(INO)+CIP*TST(INO)-CIP*(TLW(J)+460.)**4
      CALL SIMULT(A,B,2,IERROR)
      TLF(J)=B(1)
      TLW(J)=B(2)
      C9=B(1)
      C12=B(2)
 3300 IF(N.EQ.2)GO TO 3400
      J=J+IS
C     CALCULATING INNER SEG.
      A(1,1)=A9
      A(1,2)=-B9
      A(2,1)=-B9
      A(2,2)=A2
      B(1)=SPMAF(INO)*TLF(J)/DELT+(R1+C1)*TLF(J-IS)
      B(2)=SPMAW(INO)*TLW(J)/DELT+R3*TLW(J-IS)+R4*TLW(J+IS)
     + +C3*TA(INO)+CIP*TST(INO)-CIP*(TLW(J)+460.)**4
      CALL SIMULT(A,B,2,IERROR)
      TLF(J)=B(1)
      TLW(J)=B(2)
      N=N-1
      GO TO 3300
C     CALCULATING LAST NODE
 3400 CONTINUE
      J=J+IS
      R4=1.0/(2.0*DXF(INAD)/(ACW(INAD)*CW(INO)))
      A(1,1)=A9
      A(2,1)=-B9
      A(1,2)=-B9
      A(2,2)=A2
      B(1)=SPMAF(INO)*TLF(J)/DELT+(R1+C1)*TLF(J-IS)
      B(2)=SPMAW(INO)*TLW(J)/DELT+R3*TLW(J-IS)+R4*TC(IL+IS)
     + +C3*TA(INO)+CIP*TST(INO)-CIP*(TLW(J)+460.)**4
      CALL SIMULT(A,B,2,IERROR)
      TLF(J)=B(1)
      TLW(J)=B(2)
      IF(Q(IL).LT.0.0) GO TO 3500
C     TF(IL+IS)=TLF(M)
      TF(IL)=TLF(J)
C     TW(IL+IS)=TLW(M)
      TW(IL)=TLW(J)
      TF(IL+IS)=C9
      TW(IL+IS)=C12
      GO TO 3599
 3500 TF(IL+IS)=TLF(J)
C     TF(IL)=TLF(M)
      TW(IL+IS)=TLW(J)
C     TW(IL)=TLW(M)
      TF(IL)=C9
      TW(IL)=C12
 3599 CONTINUE
      N=NLSEG(INO)
```

LISTING (Continued)

```
      M=LSTART(INO)
      J=M
      JJ=M+N-1
C     WRITE(6,998)INO,M,N
  998 FORMAT(20X,//,30X,3I10)
C     WRITE(6,999)(TLF(MMM),MMM=J,JJ)
  999 FORMAT(1X,10F12.5)
 3550 CONTINUE
      RETURN
      END
```

APPENDIX V
CONTROL DATA
HYTTHA USER MANUAL (AFAPL-TR-76-43, VOL. VII)

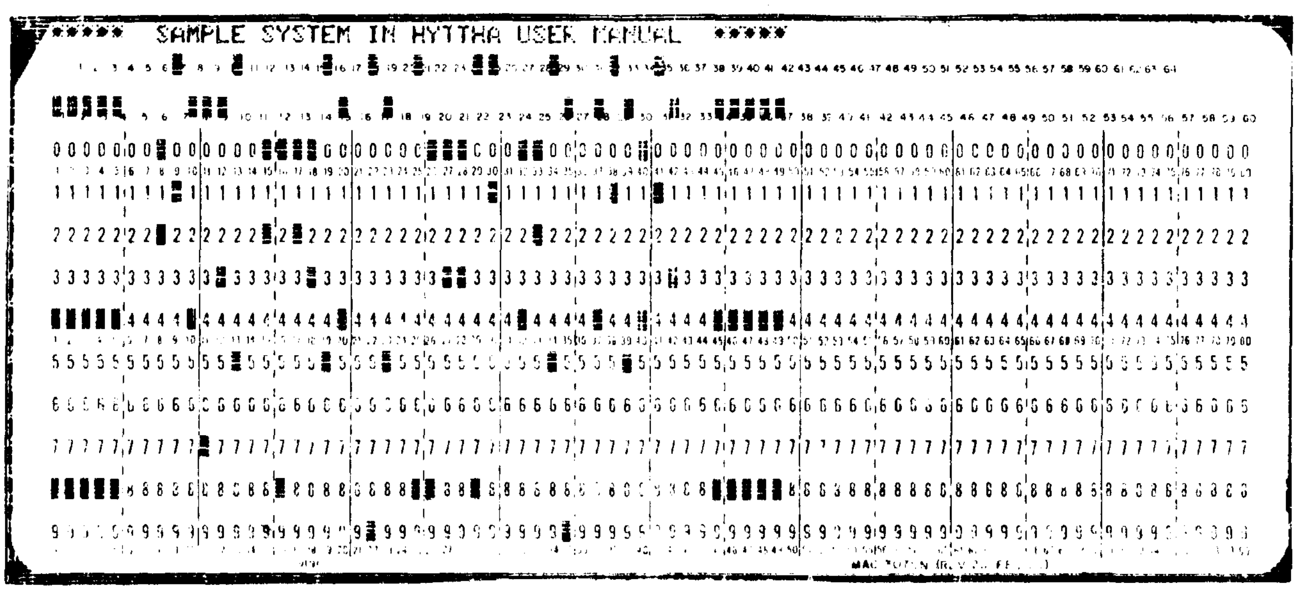
4.0 CONTROL DATA

4.1 GENERAL CONTROL DATA

This group includes three cards which set up the program title, time intervals, fluid temperature and type, number of lines and components and pressures. In addition component temperature and environment cards can be specified.

Card 1 - This card inputs the program title. A maximum of 80 alphameric characters can be used in the title starting at card column 1.

Example Card:



Card 2 - This card inputs data for the number of lines and components, and three times. Time one is the calculation time interval used as the main program time step. The second time is the total calculation time until program stops, and time three is the plotting time interval.

The time step should not be so large as to let the fluid travel further than the segment length, DELTAX during that time. If the time is chosen too large, the line routine will recalculate the segment length.

$$DELTAX = \frac{\text{flow * time step}}{\text{fluid cross sectional area}}$$

The plotting time interval is selected to suit the output device, the minimum being the calculation time interval. The actual value is usually chosen to give 101 plotted points (i.e. = final time ÷ 100 or N times the calculation time interval so that every Nth calculated point is plotted).

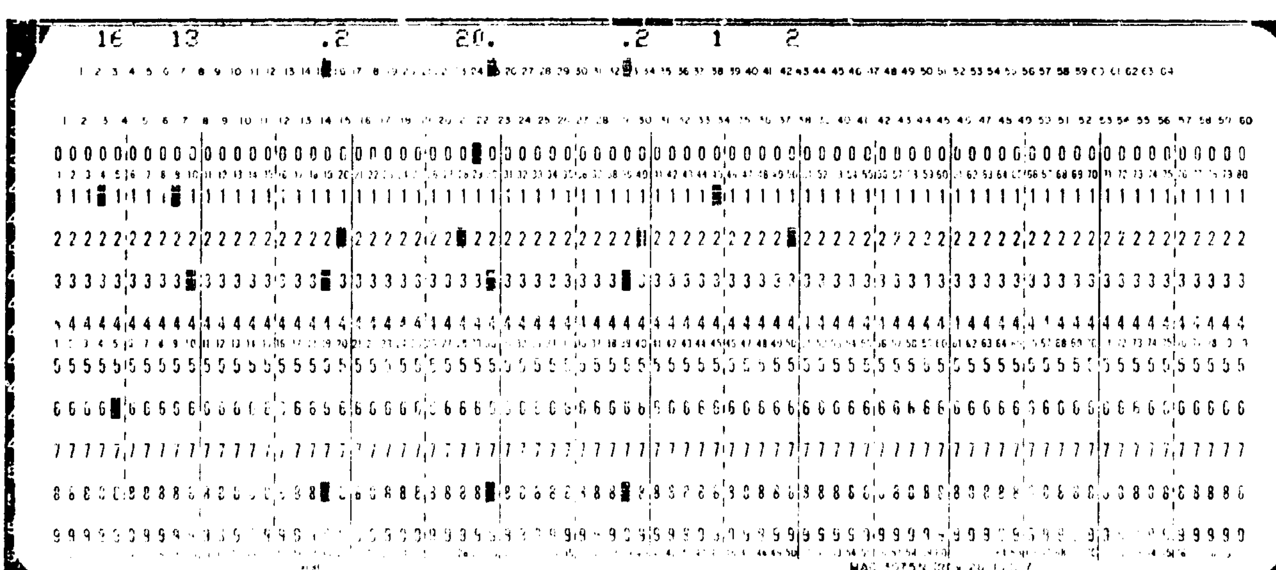
Every component requires the user to specify the ambient, structural, wall, and fluid temperatures. Specifying a one in column 45 will cause the reading of an extra data card containing these default temperature values.

The user can input different operating environments. The number of environments is written in column 50.

CARD NUMBER 2

COLUMN	FORMAT	DATA	DIMENSIONS
1-5	I5	Total Number of Lines	—
6-10	I5	Total Number of Components	—
11-20	E10.0	Calculation Time Interval, DELT	SEC
21-30	E10.0	Final Time	SEC
31-40	E10.0	Plotting Time Interval	SEC
41-45	I5	Temperature Indicator 0 = uses element temp 1 = uses default temp	--
46-50	I5	Number of Environments	

EXAMPLE CARD



Card 2a - Component Temperature Option

If the user puts a 1 in column 45 of card 2, the HYTTHA program will read an extra card. The data card contains indicator and initial default temperatures for the atmosphere, structure, wall and fluid. When a particular indicator is selected, the appropriate default temperature will be used by all the components including the lines. However each component has an override on the integer data card which will suppress using the default values.

Card 2a is intended for use whenever the programmer wishes to avoid entering temperature data for all the components. Component temperatures can still be specified.

Card 2b - Environment Option

Initial ambient, structural, and wall temperatures should be entered for the number of environments in column 50 of card 2. Currently five different environments can be input. The individual components must specify which environment it will use. This section will override the temperatures specified on card 2a. However, the user still has the option to change the temperature data for each component.

COMPONENT TEMPERATURE OPTION

CARD NUMBER 2a

COLUMN	FORMAT	DATA	DIMENSIONS
1-5	I5	Ambient Temperature Indicator — 0 = use component values; 1 = use default value	--
6-15	E10.0	Default Ambient Temperature	°F
16-20	I5	Structural Temperature Indicator — 0 = use component value; 1 = use default value	--
21-30	E10.0	Default Structural Temperature	°F
31-35	I5	Wall Temperature Indicator — 0 = use component value; 1 = use default value	--
36-45	E10.0	Default Wall Temperature	°F
46-50	I5	Fluid Temperature Indicator — 0 = use component value; 1 = use default value	--
51-60	E10.0	Default Fluid Temperature	°F

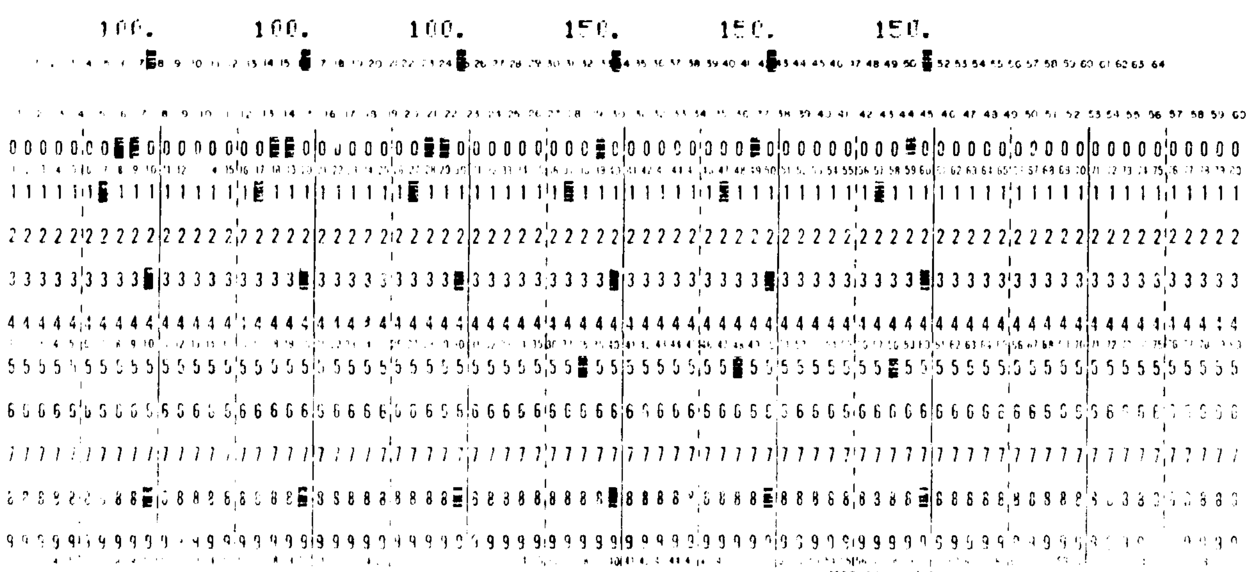
EXAMPLE CARD

ENVIRONMENT OPTION

CARD NUMBER 2b

COLUMN	FORMAT	DATA	DIMENSIONS
1-10	E10.0	Environment #1 Ambient Temperature	°F
11-20	E10.0	Environment #1 Structural Temperature	°F
21-30	E10.0	Environment #1 Wall Temperature	°F
31-40	E10.0	(Continue sequence for as many	
41-50	E10.0	environments entered on card 2	
51-60	E10.0	column 50. Five is the maximum)	
61-70	E10.0		
71-80	E10.0		

EXAMPLE CARD



<u>Card 3</u> - The first value is the fluid type. The program is set up to run with either of the following fluid types at any temperature from -65°F to 300°F. If a computed temperature exceeds 300°F, the fluid properties at 300°F are used.

Type #1 MIL-H-5606B

Type #2 MIL-H-83282

Type #3 Skydrol 500B

The fluid type number selects the fluid data to be used from tabulated data stored in the program and adjusts the fluid properties to the computed pressure.

The second value is the fluid temperature throughout the system. This is intended as a default value should the user forget to enter the fluid temperature on the component cards. The fluid temperature will default to 100°F if this column is left blank.

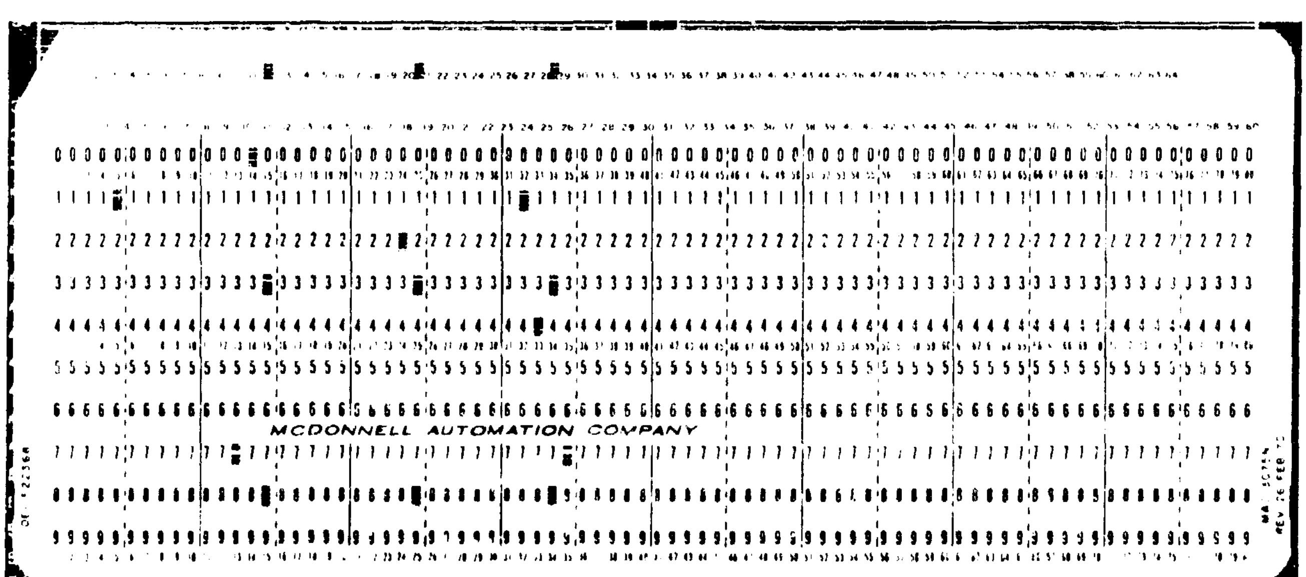
The third value is the fluid vapor pressure at the fluid temperature. Note: If the vapor pressure is not input the program will use a value of 2 psia.

The last value is the atmospheric pressure at the conditions of the run. The valve 14.7, atmospheric pressure at sea level will be used if this value is not input.

CARD NUMBER 3

COLUMN	FORMAT	DATA	DIMENSIONS
1-5	I5	Fluid Type Number	
6-15	E10.0	Fluid Temperature	°F
16-25	E10.0	Fluid Vapor Pressure	PSI
26-35	E10.0	Atmospheric Pressure	PSI

EXAMPLE CARD



APPENDIX V
HYTTHA OUTPUT OPTION
HYTTHA USER MANUAL (AFAPL-TR-76-43, VOL. VII)

7.0 SYSTEM ARRANGEMENT DATA

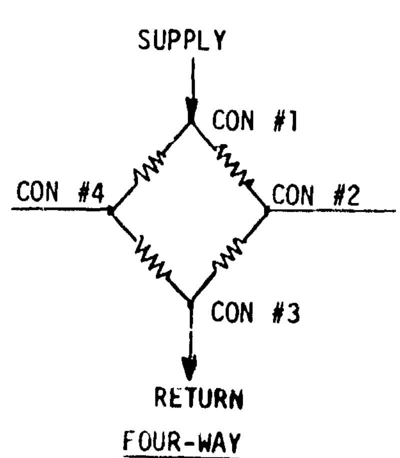
This section of the input data is used to describe the system arrangement. Having input the necessary information for all the system lines and components, one must now input the way in which these components and lines are interconnected.

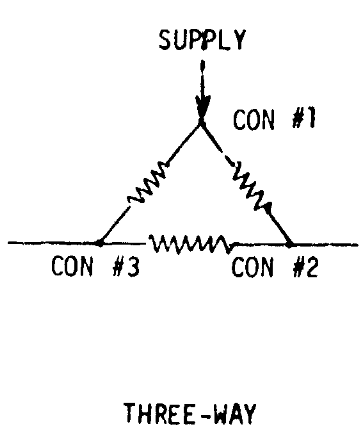
Special Cases

If a leg is terminated by a constant pressure source, the constant pressure has to be input along with the leg connection information via a type 61 reservoir. A current restriction requires that only nodes with a single leg can have a constant pressure termination. A second restriction is that there must be at least one variable node. Nodes should not be placed in the center of any component having a pressure loss since each leg connected to the node will include the pressure drop of the component.

Other component restrictions are as follows:

VALVES - TVALV22 can require three or four nodes depending upon the valve usage. The four-way valve and three-way versions of TVALV22 are described as follows:





FOUR WAY

THREE WAY

The valve schematic should be established for steady state operation including any interflow paths. A node is then required at every connection that splits or merges flow (including interflow leakage) and at any connection that terminates flow.

Actuators - Unbalanced actuators must include a node which is used to account for any flow gain or loss in event the actuator is in motion during steady state conditions.

Reservoirs -

o RSVR61 requires one node

o RSVR62 requires two nodes open ended (not connected by a leg). One node is considered to be on the low pressure side with the other node considered on the high pressure side.

7.1 GENERAL DATA

On this card input the number of nodes, the number of legs and steady state printing options.

0 - print node pressures and leg flows for each steady state iteration

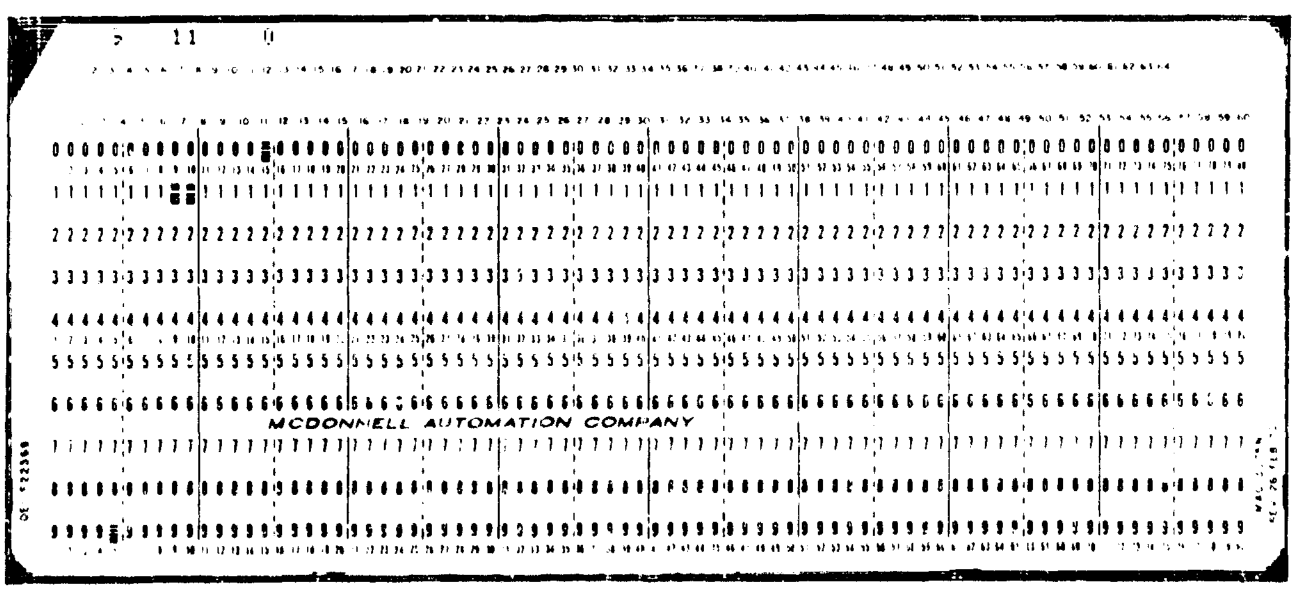
1 - no steady state data is printed

2 - print balanced node pressures and leg flows at each time step

CARD NUMBER 1

COLUMN	FORMAT	DATA
1-5	I5	Number of Nodes
6-10	I5	Number of Legs
11-15	I5	Steadv State Printing Options 0, 1, or 2
16-20	I5	
21-25	I5	
26-30	I5	
31-35	I5	
36-40	I5	
41-45	I5	
46-50	I5	
51-55	I5	
56-60	I5	
61-65	I5	
66-70	I5	
71-75	I5	
76-80	I5	

EXAMPLE CARD



8.0 <u>OUTPUT REQUIREMENTS DATA</u>

The program will output in a print plot form, any calculated system variable versus time. The time interval between plotted points is input on the first general control card.

When using the print plot routine, it should be noted that 101 points are the maximum that can be plotted on one page. When more than 101 points are requested, the plot is continued on an additional page(s).

On the first output data card the user specifies the number of line plot data cards, the number of plotted component variables, and specialized outputs.

The line variables which can be selected are the pressures, flows, wall temperature, and fluid temperature up and downstream of each line. The component variables which can be selected are listed in paragraph 8.2.

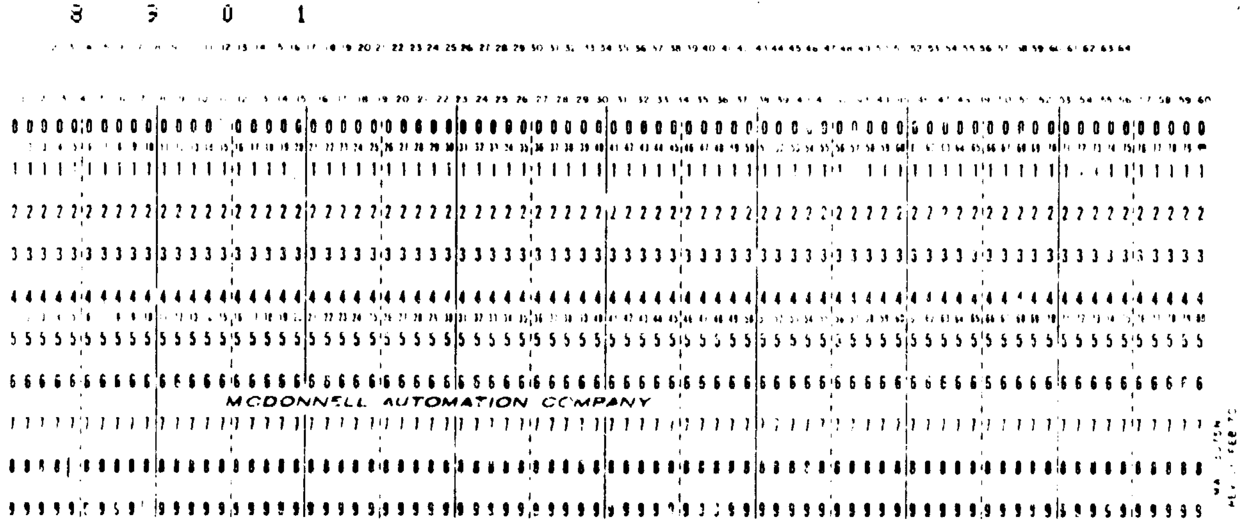
The Y axis scale can be selected for any print plot. The scale data card is explained in Paragraph 8.3. This card follows the component variable cards.

PLOT DATA CARD

COLUMN	FORMAT	DATA
1-5	I5	Number of line plot data cards
6-10	I5	Number of component variables to be plotted
11-15	I5	0
16-20	I5	0 - Normal graphs +1 - Prints listing of plotted values
21-25	I5	0 - Normal graphs +1 - No graphs
26-30	I5	0 - Normal graphs +1 - Prints magnitudes (*)
31-35	I5	Number of scales

EXAMPLE CARD

8 9 0 1



8.3 Selecting Output Plot Scales

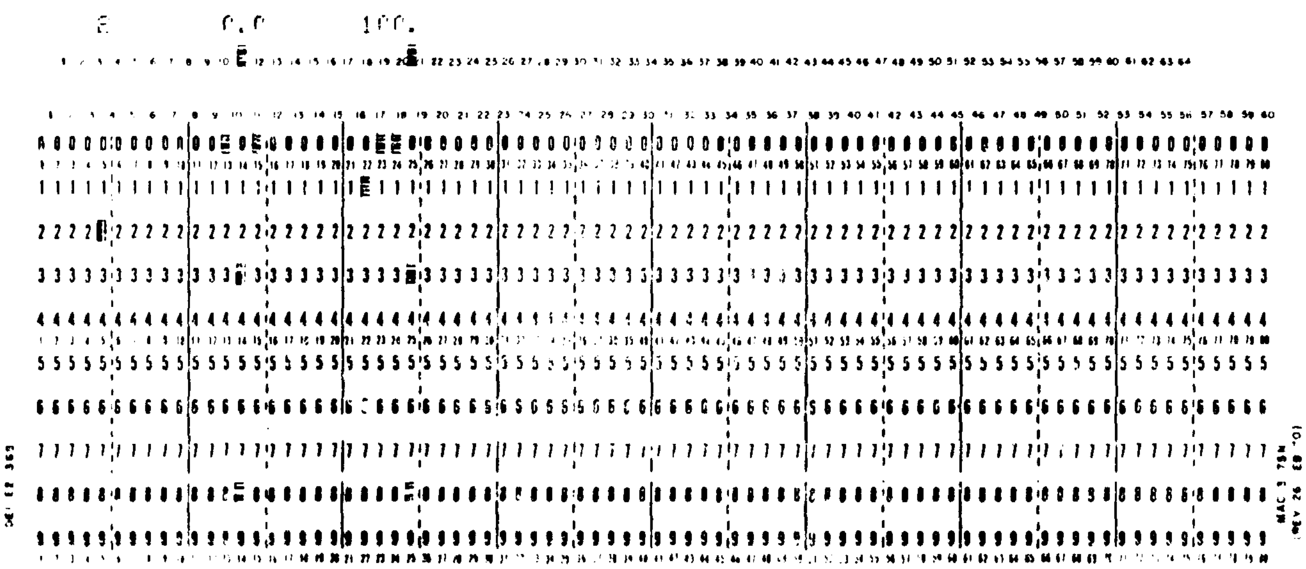
The user has the option of selecting the Y axis scale for any output plot. The scale will remain the same even if the plot is continued on an additional page(s). The graph number must be specified followed by the minimum and maximum Y values as listed on the scale data card. The graph numbering is started in sequence from the first plot entry on the first line data card or the first component variable if there are no line data plots.

Note: When using this option, the maximum Y value cannot be 0.0. If Y max is set to zero the output routine will automatically generate the scale.

SCALE DATA CARD

COLUMN	FORMAT	DATA	DIMENSIONS
1-5	I5	Graph number	
6-15	F10.0	Minimum Y Value	
16-25	F10.0	Maximum Y Value	
		Repeat sequence using additional cards	
		(if necessary) until the number of graphs	
		in columns 41-45 on plot data card is	
		reached.	

EXAMPLE CARD



APPENDIX W
THE BMOMENT PROGRAM
LIST OF VARIABLES

AREAT	AREA OF TRIANGLE IN CALCULATION OF MSG( )(SQ.IN.)
A( ), B( ), C( ), D( ), E( )	ARRAY OF MEASUREMENTS (IN) SEE FIGURE A-1
CP	CRITICAL POINT IN ΔL SECTION
CPE( )	CRITICAL POINT STRAIN (MICRO IN/IN)
CPS( )	CRITICAL POINT STRESS (PSI)
EP( )	ARRAY OF STRAINS AT SG( ) (MICRO IN/IN)
EPS( )	ARRAY OF STRAINS AT A( ) (MICRO IN/IN)
FREQ	FREQUENCY (HZ)
I,J,JJ,K, KK,N	INTEGERS (N.D.)
L	LENGTH OF BEAM (IN)
LCP	LOCATION OF CRITICAL POINT ON BEAM (IN.)
LR	VERTICAL REACTION AT LEFT END OF ΔL SECTION (LB)
M1	MOMENT AT LEFT END OF ΔL SECTION (LB-IN)
M( )	ARRAY OF BENDING MOMENTS (LB-IN)
MSG( )	BENDING MOMENT AT SELECTED LOCATIONS (LB-IN)
NCP	NUMBER OF CRITICAL POINTS (N.D.)
PSPEED	PUMP SPEED (RPM)
RR	TOTAL RIGHT REACTION (LB)
S( )	ARRAY OF STRESS AT SG( ) (PSI)
SG( )	ARRAY OF SELECTED LOCATIONS (IN.)
SIG( )	ARRAY OF STRESSES AT A( ) (PSI)
SLOPE	SLOPE OF ΔL SECTION IN SHEARING DIAGRAM (N.D.)
TLR	VERTICAL REACTION AT LEFT END (LB)
TM1	MOMENT AT LEFT END OF BEAM (LB-IN)
TM2	TOTAL MOMENT AT RIGHT END (LB-IN)
V( )	ARRAY OF SHEARING FORCES (LB)
VSG( )	SHEAR FORCE AT SELECTED LOCATIONS (LB)
VSQ	VALUE OF RADICAND
VSQR	VALUE OF SQUARE ROOT IN QUADRATIC EQUATION
W( )	ARRAY OF LOADS (LB/IN)
XINT( )	ARRAY OF LOCATIONS WHERE BENDING MOMENT EQUALS ZERO (IN.)

APPENDIX W

The program for the calculation of reactions, shears, and moments for a beam with built-in ends, Figure W-1, with variations in load distribution is given in Figure W-2. Examples applied to the unclamped test specimen are shown in Figures W-3 through W-5 for one mechanical resonance at 3440 rpm and two hydraulic resonances at 2900 rpm and 4350 rpm, respectively.

A computer printout for the case at 4350 rpm is as follows:

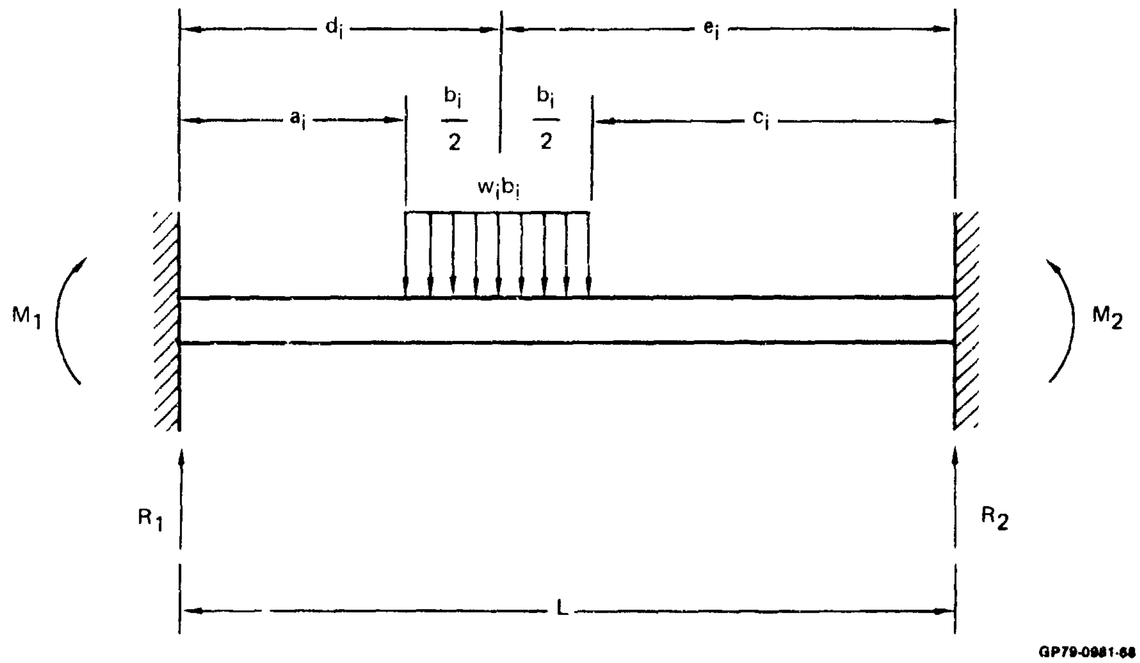


FIGURE W-1 **BEAM WITH BUILT-IN ENDS COMBINED WITH VARIATIONS IN UNIFORM LOAD DISTRIBUTION**

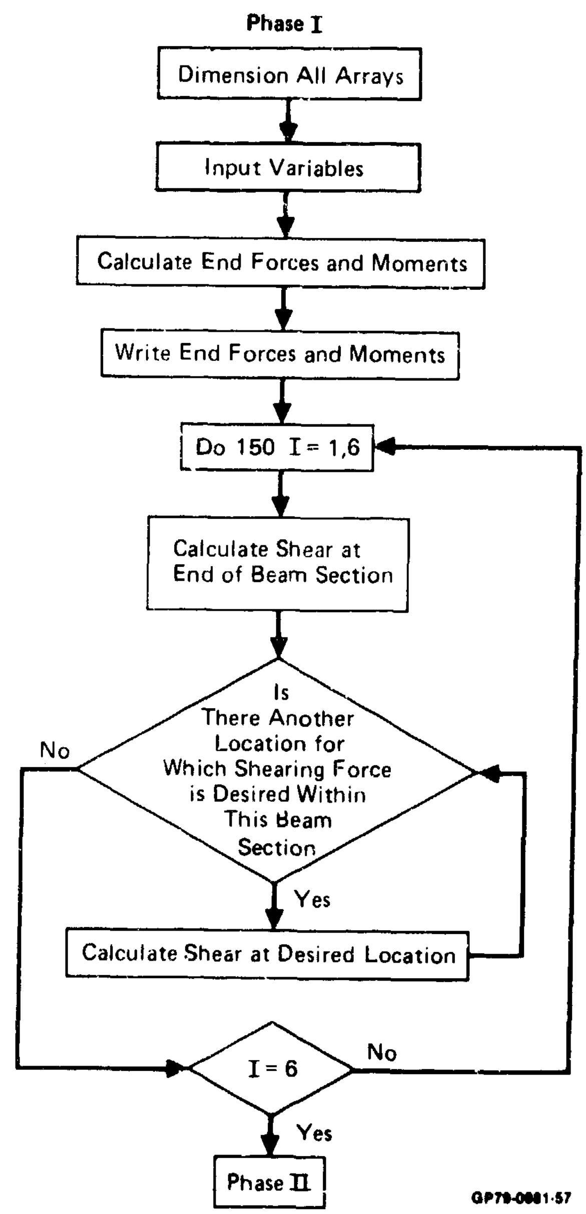


FIGURE W-2 COMPUTER PROGRAM FLOW CHART
Beam with Built-In Ends Combined with Variations in Uniform Load Distribution

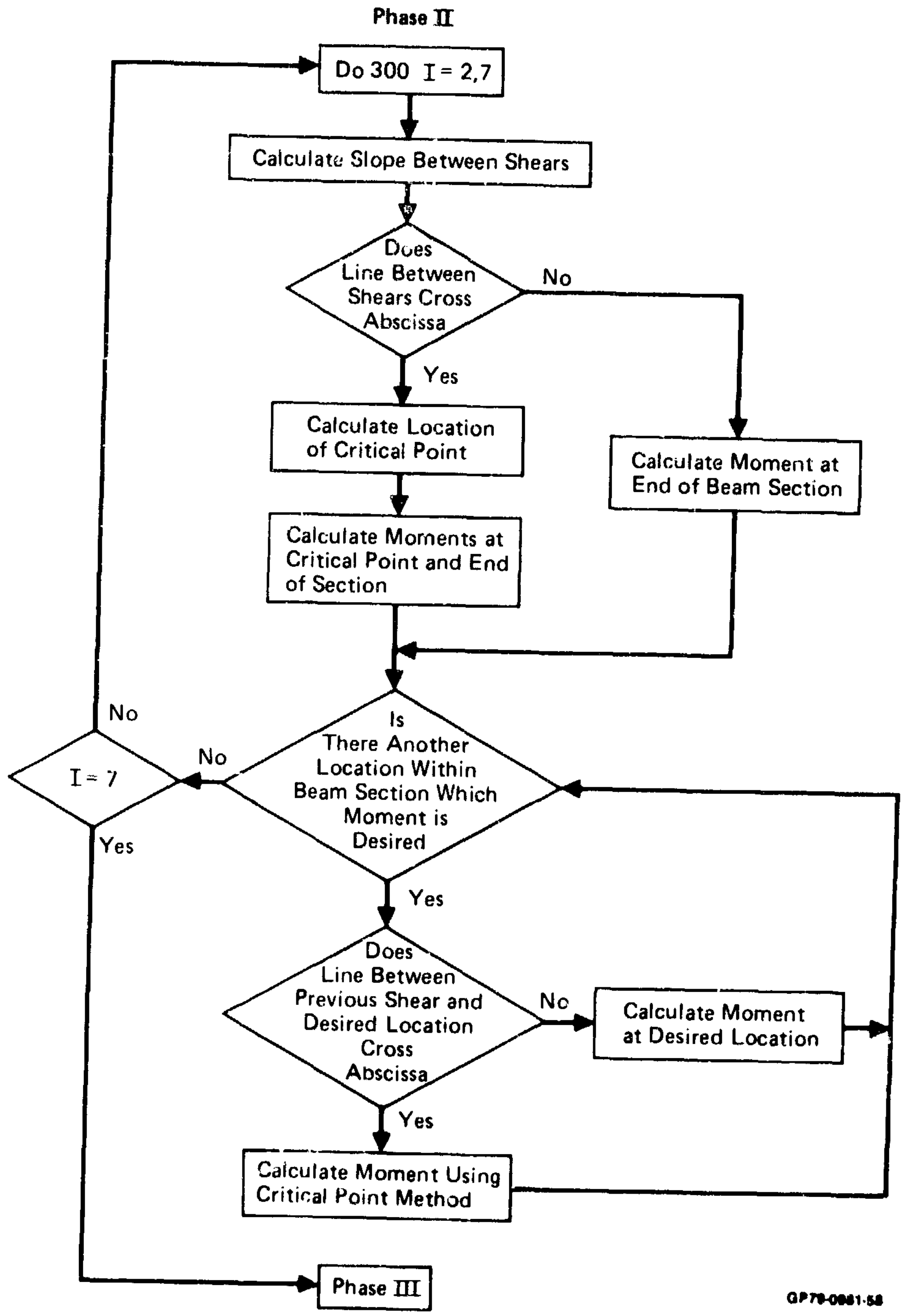


FIGURE W-2 (CONTINUED)

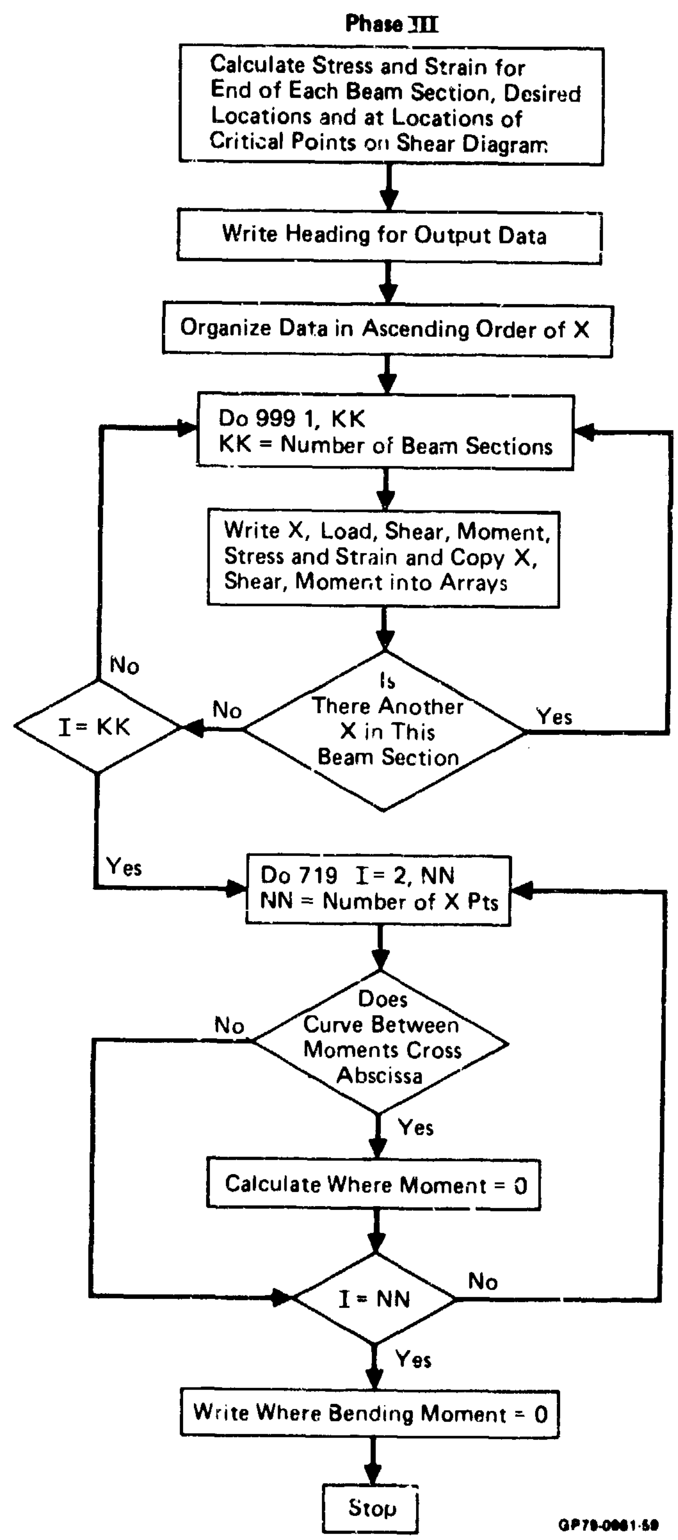


FIGURE W-2 (CONTINUED)

BMOMENT

```
      DIMENSION V(7),CP(5),W(6),A(7),B(6) C(6),D(6),E(6)
      DIMENSION SG(7),XINT(7),VSG(7),SIG(7),EPS(7),S(7)
      DIMENSION EP(7),CPS(5),CPE(5),SHEAR(25),BM(25),X(25)
      REAL L,LR,M(12),M1,LCP(5).MSG(7)
      DATA(SG(J),J=1.7)/2.58,8.7.17.7,26.6,35.5,44.5.53.2/
      DATA FREQ,PSPEED/653,4350/
      DATA(W(I),I=1,6)/2.1,5.71,-7.67,3.21,-14.04,-8.03/
      DATA(A(I),I=1,7)/0.,5.64 13.2,22.15,31.05,40.,53.2/
      DATA(B(I),I=1,6)/5.64,7 56,8.95,8.9,8.95,13.2/
      DATA(C(I),I=1,6)/47.56,40. 31.05,22.15,13.2,0./
      DATA(D(I),I=1,6)/2.82,9.42,17.675,26.6,35.525,46.6/
      DATA(E(I),I=1,6)/50.38,43.78,35.525,26.6,17.675,6.6/
      WRITE(6,1)
    1 FORMAT(16HONE - ELBOW PIPE/3X,9HUNCLAMPED//)
      WRITE(6,2)FREQ.PSPEED
    2 FORMAT(4X,9HFREQUENCY,3X,I4/4X,10HPUMP SPEED,3X,I4)
      WRITE(6,3)
    3 FORMAT(4X 7HINPLANE///)
      WB=0.
      WBE=0.
      TM1=0.
      TLR=0.
      L=53.2
```

INPUT DATA FOR ONE-ELBOW PIPE TEST SPECIMEN

```
      DO 50 I=1,6
      WB=W(I)*B(I)*(-1.)+WB
      WBE=W(I)*B(I)*E(I)*(-1.)+WBE
      C1=W(I)*B(I)*(-1.)/(4 *L**3)
      P1=4.*E(I)**2*(L+2.*D(I))-B(I)**2*(C(I)-A(I))
      LR=C1*P1
      C2=W(I)*B(I)*(-1.)/(24.*L**2)
      P2=B(I)**2*(L+3.*(C(I)-A(I)))-24.*E(I)**2*D(I)
      M1=C2*P2
      TM1=M1+TM1
      TLR=LR+TLR
   50 CONTINUE
      RR=WB-TLR
      TM2=TLR*L-WBE+TM1
      WRITE(6,75)TLR RR TM1.TM2
```

DETERMINATION OF END REACTIONS AND BUILT-IN MOMENTS

```
   75 FORMAT(4X,3HR1=,F9.3,9X,3HR2=,F9.3/4X,3HM1=,F9.3,9X,3HM2=,F9.3)
      V(1)=TLR
      M(1)=TM1
      J=1
      DO 150 I=1 6
      V(I+1)=W(I)*B(I)+V(I)
   90 IF(SG(J).GT.A(I+1)) GOTO 150
      VSG(J)=W(I)*(SG(J)-A(I))+V(I)
      J=J+1
      IF(J.EQ.7) GOTO 150
      GOTO 90
  150 CONTINUE
```

DETERMINATION OF SHEAR LOADS ALONG SPECIMEN

```
      NCP=0
      NCPSG=0
      J=1
      DO 300 I=2,7
      SLOPE=(V(I)-V(I-1))/B(I-1)
      IF(V(I).GT.0.) GOTO 200
      IF(V(I-1).LT.0.) GOTO 250
      GOTO 225
  200 IF(V(I-1).GT.0 ) GOTO 250
  225 NCP=NCP+1
      CP(NCP)=ABS(V(I-1)/SLOPE)
      LCP(NCP)=CP(NCP)+A(I-1)
      M(7+NCP)=.5*CP(NCP)*V(I-1)+M(I-1)
      M(I)=.5*CP(NCP)*V(I-1)+.5*(B(I-1)-CP(NCP))*V(I)+M(I-1)
      GOTO 283
  250 M(I)=V(I-1)*B(I-1)+.5*(V(I)-V(I-1))*B(I-1)+M(I-1)
  283 IF(SG(J).GT.A(I)) GOTO 300
      IF(VSG(J).GT.0.) GOTO 280
      IF(V(I-1).LT.0.) GOTO 298
      GOTO 281
  280 IF(V(I-1).GT.0.) GOTO 298
  281 NCPSG=NCPSG+1
      MSG(J)=.5*CP(NCP)*V(I-1)+.5*(SG(J)-A(I-1)-CP(NCP))*VSG(J)+M(I-1)
      GOTO 299
  298 AREAT=.5*(VSG(J)-V(I-1))*(SG(J)-A(I-1))
      MSG(J)=AREAT+V(I-1)*(SG(J)-A(I-1))+M(I-1)
  299 J=J+1
      IF(J.EQ.7) GOTO 300
      GOTO 283
  300 CONTINUE
```

COMPUTATION OF BENDING MOMENTS AND CRITICAL POINTS (WHERE SHEAR = 0)

```
      DO 301 I=1,7
      SIG(I)=29.13*M(I)
      EPS(I)=1.765*M(I)
  301 CONTINUE
      DO 303 I=1,6
      S(I)=29.13*MSG(I)
      EP(I)=1.765*MSG(I)
  303 CONTINUE
      DO 700 I=1,NCP
      CPS(I)=29.13*M(7+I)
      CPE(I)=1.765*M(7+I)
  700 CONTINUE
```

COMPUTATION OF STRESSES AND STRAINS FROM BENDING MOMENTS

```
      WRITE(6,170)
  170 FORMAT(//4X,1HX,12X 1HL,14X,1HV,11X,2HBM,7X,6HSTRESS,12X,6HSTRAIN)
      I=1
      VC=0.
      NN=1
      J=1
      K=1
      DO 999 KK=1,6
  799 IF(A(I).LE.SG(J)) GOTO 805
      IF(K.GT.NCP) GOTO 815
      IF(SG(J).LE.LCP(K)) GOTO 815
  800 WRITE(6,176)LCP(K),W(KK),VC,M(7+K),CPS(K),CPE(K)
      SHEAR(NN)=VC
      BM(NN)=M(7+K)
      X(NN)=LCP(K)
      NN=NN+1
      K=K+1
      GO TO 998
```

ORGANIZATION OF OUTPUT DATA

ORGANIZATION OF OUTPUT DATA (CONTINUED)

```
805 IF(K.GT.NCP) GOTO 810
    IF(A(I).LE.LCP(K)) GOTO 810
    GOTO 800
810 WRITE(6,175)A(I),W(KK),V(I),M(I),SIG(I),EPS(I)
    SHEAR(NN)=V(I)
    BM(NN)=M(I)
    X(NN)=A(I)
    NN=NN+1
    I=I+1
    GOTO 998
815 WRITE(6,177)SG(J) W(KK) VSG(J),MSG(J),S(J),EP(J)
    SHEAR(NN)=VSG(J)
    BM(NN)=MSG(J)
    X(NN)=SG(J)
    NN=NN+1
    J=J+1
998 IF(I.EQ.8) GOTO 999
    IF(A(KK+1).EQ.0.)A(KK+1)=53 2
    IF(SG(J).LE.A(KK+1)) GOTO799
    IF(K.GT.NCP) GOTO 999
    IF(LCP(K).LE.A(KK+1)) GOTO799
175 FORMAT(1X,F6.2,6X,F7.3,7X,F9.3,3X,F9.3,3X,F9.2,8X,F9.2/)
176 FORMAT(1H#,F6.2,6X,F7.3,7X,F9.3,3X,F9.3,3X,F9.2,8X,F9.2/)
177 FORMAT(1H*,F6.2,6X,F7.3,7X,F9.3,3X,F9.3,3X,F9.2,8X,F9.2/)
999 CONTINUE
```

COMPUTATION OF LOCATIONS WHERE BENDING MOMENT VALUES EQUAL ZERO

```
    JJ=1
    DO 719 I=2,NN
    SLOPE=(SHEAR(I)-SHEAR(I-1))/(X(I)-X(I-1))
275 IF(BM(I).GT.0.)GOTO 254
    IF(BM(I-1).LT.0.) GOTO 719
    GOTO 255
254 IF(BM(I-1).GT.0.)GOTO 719
255 VSQ=SHEAR(I-1)**2-4.*.5*SLOPE*BM(I-1)
    IF(VSQ.LT.0.) GOTO 719
    VSQR=SQRT(VSQ)
    XINT(JJ)=((-1.)*SHEAR(I-1)-VSQR)/SLOPE+X(I-1)
    IF(XINT(JJ).GT.X(I)) GOTO 269
268 IF(XINT(JJ).GT.X(I-1)) GOTO 270
269 XINT(JJ)=((-1.)*SHEAR(I-1)+VSQR)/SLOPE+X(I-1)
270 JJ=JJ+1
719 CONTINUE
    WRITE(6,181)
181 FORMAT(//22H#  CRITICAL POINT DATA)
    WRITE(6,182)
182 FORMAT(19H*  STRAIN GAGE DATA//)
    N=JJ-1
    WRITE(6,190)(XINT(I),I=1,N)
190 FORMAT(32HBENDING MOMENT EQUALS ZERO AT X=,F6.2/)
    END
```

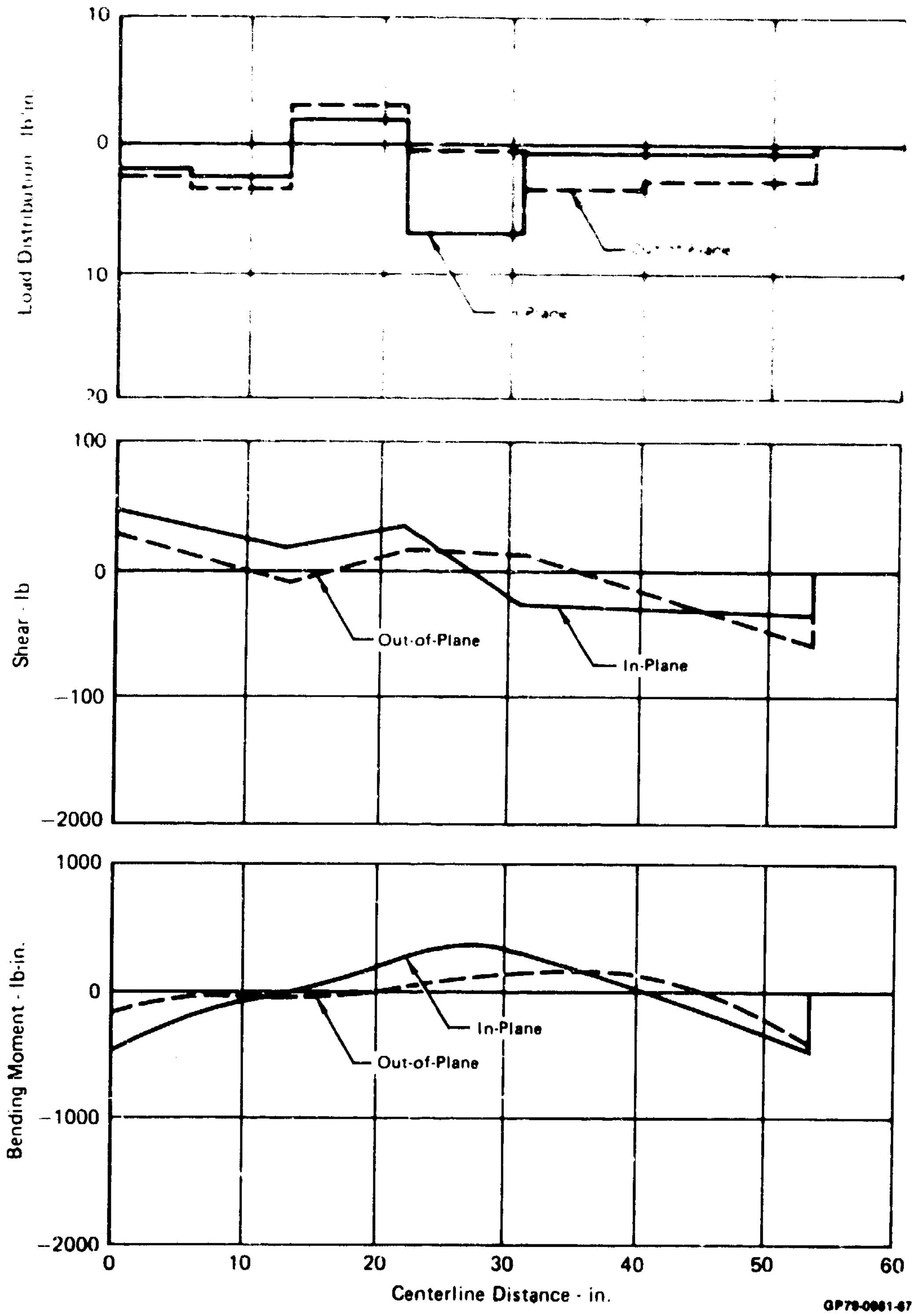


FIGURE W-3 ONE-ELBOW PIPE UNCLAMPED STRUCTURAL ANALYSIS
Frequency 1030 Hz Pump Speed 3440 RPM

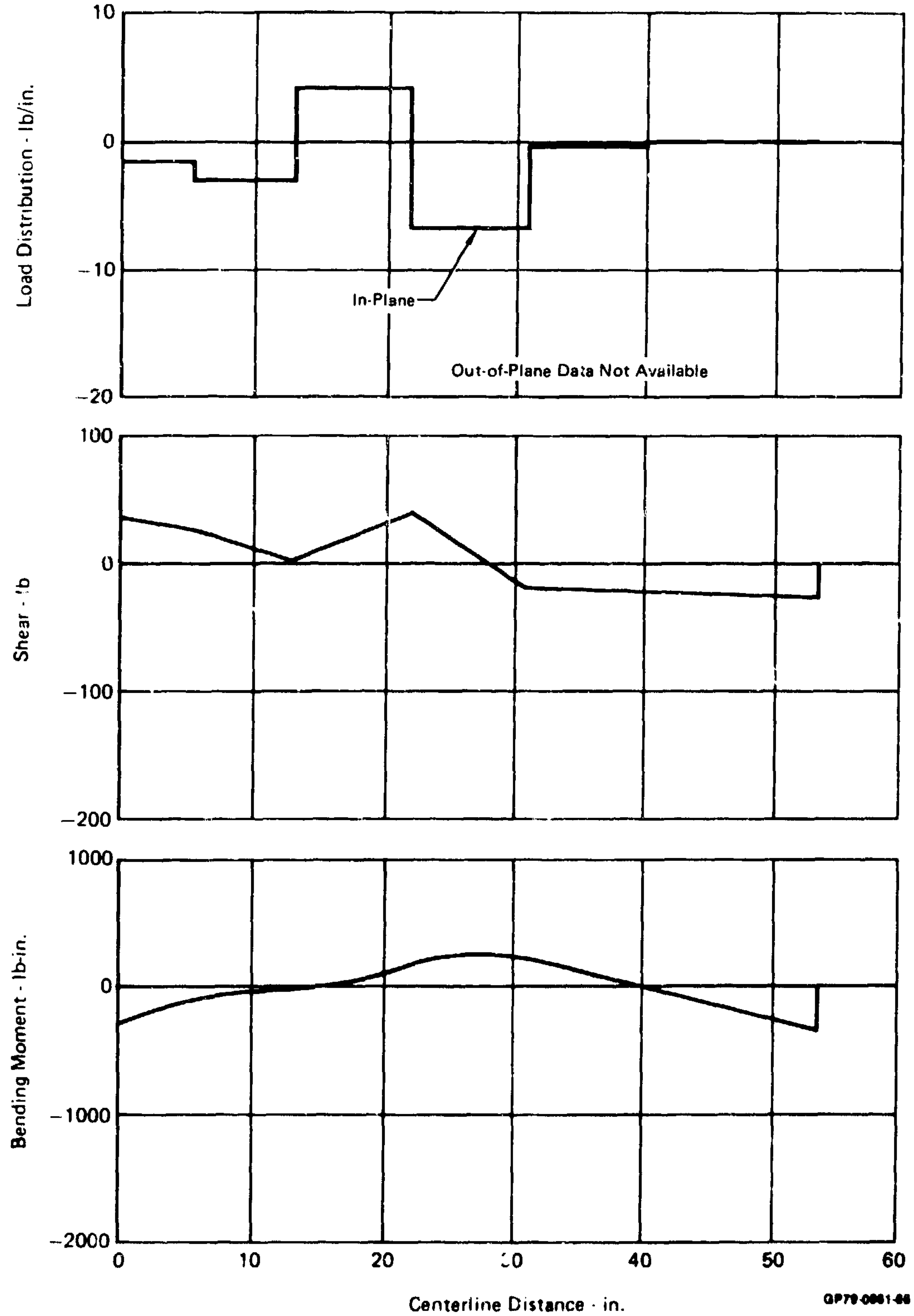


FIGURE W-4 ONE-ELBOW PIPE UNCLAMPED STRUCTURAL ANALYSIS
Frequency 870 Hz Pump Speed 2900 RPM

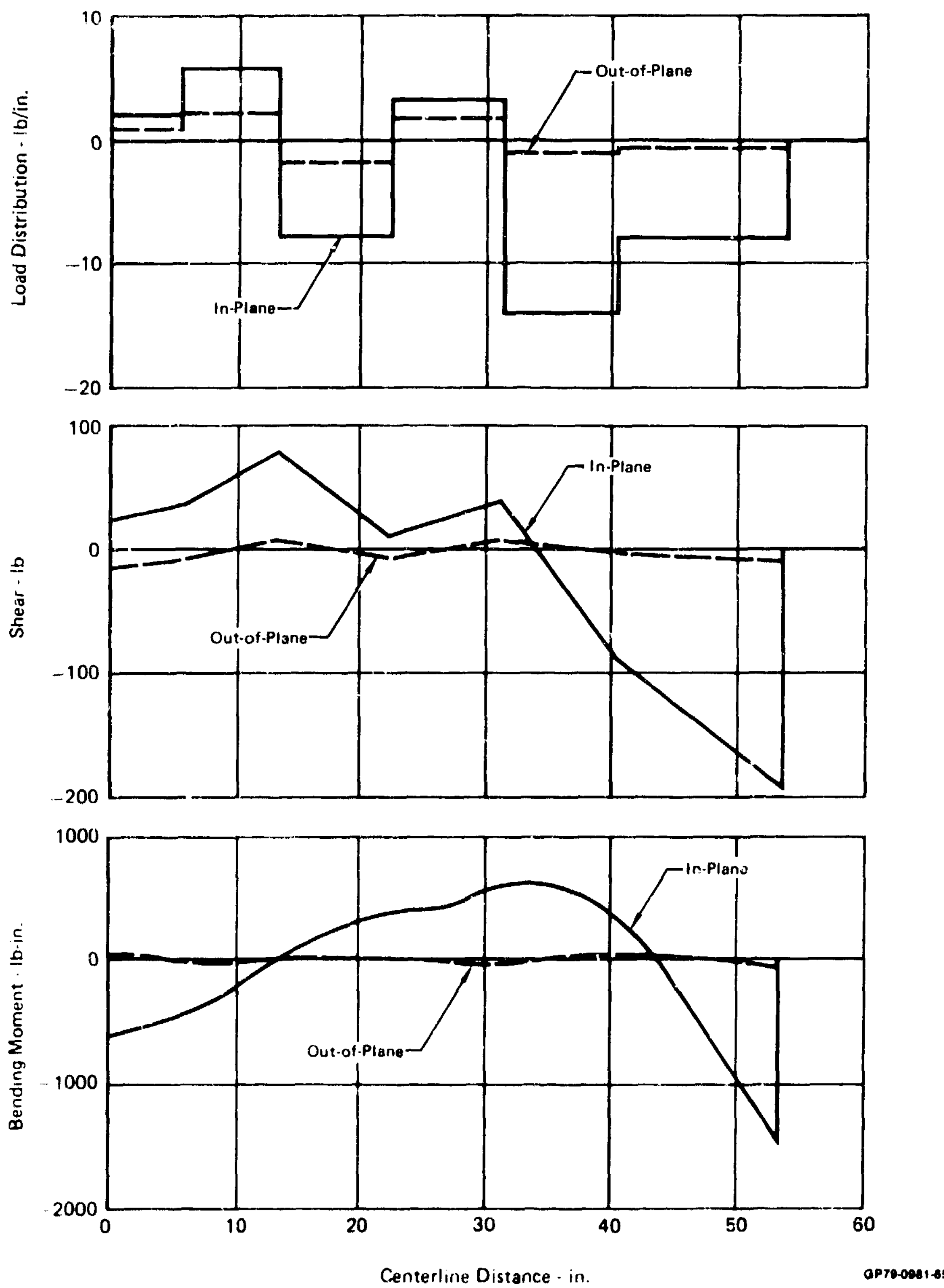


FIGURE W-5 ONE-ELBOW PIPE UNCLAMPED STRUCTURAL ANALYSIS
Frequency 653 Hz Pump Speed 4350 RPM

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